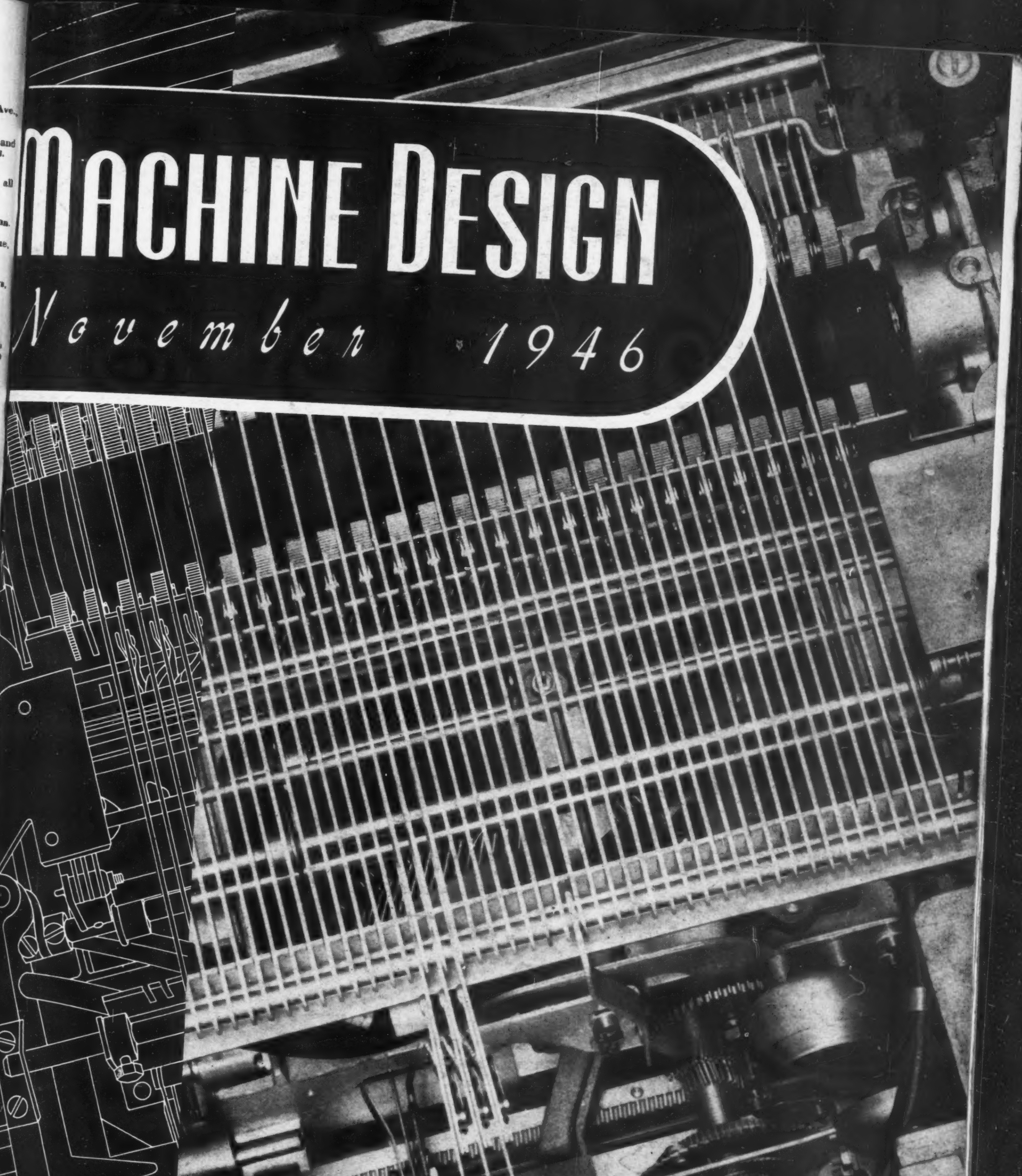
MACHINE DESIGN

November 1946

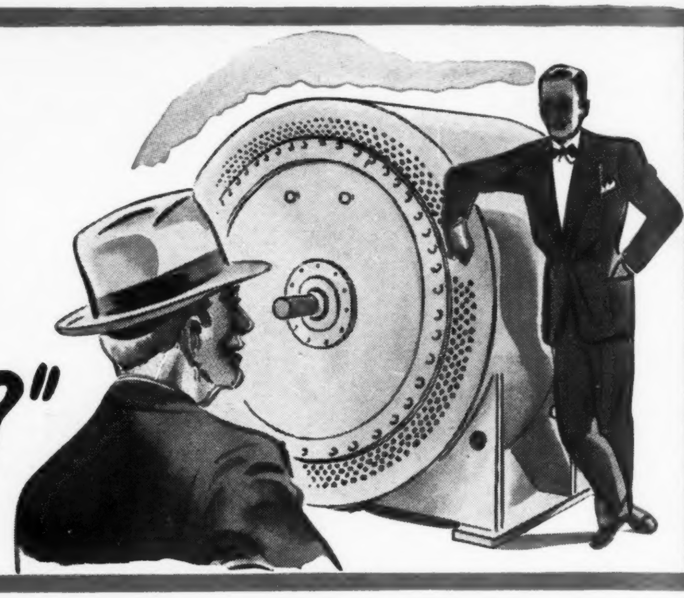


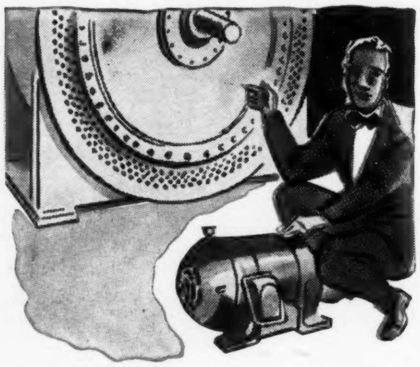
In This Issue:

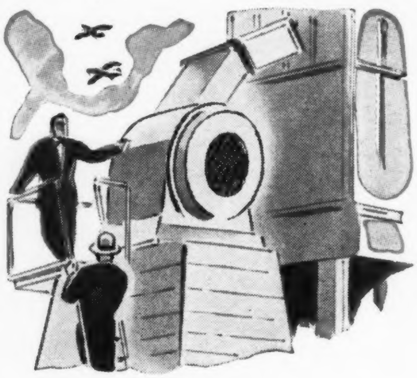
Engineering Record Systems

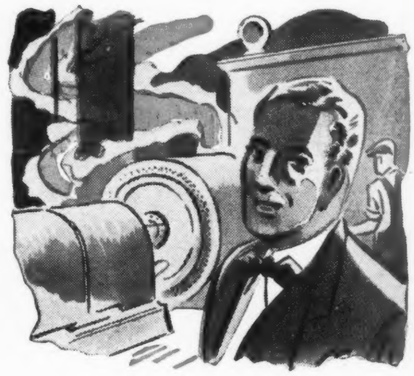
Planetary Transmissions

"Did You Say, Up to 2000 hp ...Weatherproof?"









"Right. In Allis-Chalmers' modern motor line we're now prepared to offer you a *complete range* of outdoor weatherproof, totally-enclosed motors all the way from small, standard ratings to sizes of 2,000 hp or larger."

"Why, That Means Then," our customer exclaimed, "in larger ratings they fill the need of power plant engineers who want to install equipment like draft fans *outside,* and thus keep down structure costs per kw capacity." Right again —and they fit into *other* industries too!

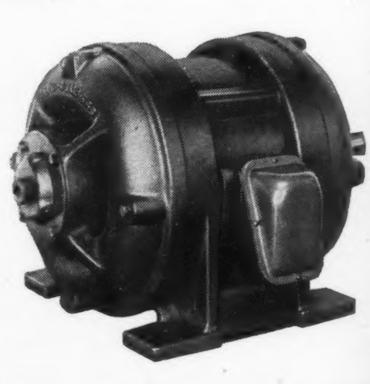
Chemical, for instance. Here cast iron motors have generally been preferred ... because they stand up better under acid and alkali conditions. But now these *new* motors, with *new* design improvements, promise to provide several important operating advantages ...

We've Re-designed the heat-transfer system to *improve ventilation.* We've *reduced maintenance* by making air passages not only more accessible, but practically *self*-cleaning as well. And we've *eliminated* liquid and dirt-trapping pockets. In the smaller ratings we of course offer our standard design.

Looking for a Good Motor?

SPECIFY "Allis-Chalmers"! For whether your job calls for a large motor like that above, or a small standard machine, you can be sure of getting two things when you specify "Allis-Chalmers:" 1) Exactly the right type and size motor; 2) consistent with the finest quality workmanship of which we are capable.

For the full story of liberal Allis-Chalmers motor construction, call our nearby dealer or sales office. ALLIS-CHALMERS, MILWAUKEE 1, WIS. A 2117



ALLIS-CHALMERS

One of the Big 3 in Electric Power Equipment — Biggest of All in Range of Industrial Products

MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

MAIN OFFICE

Penton Building, Cleveland 13, Ohio

BRANCH OFFICES

New York 17 .... 16 East 43rd St.
Chicago 11 .... 520 North Michigan Ave.
Pittsburgh 19 .... 2800 Koppers Building
Detroit 2 .... 6560 Cass Ave.
Washington 4 ... 956 National Press Bldg.
Los Angeles 4 130 N. New Hampshire Ave.
London, S.W.1 2 Caxton St., Westminster

Published by The Penton Publishing Co. E. L. Shaner, Pres. and Treas.; G. O. Hays, Vice Pres. and Gen. Mgr.; R. C. Jaenke, Vice Pres.; F. G. Steinebach, Vice Pres. and Secy.; E. L. Werner, Asst. Treas. Published on the seventh of each month.

THIS MONTH'S COVER: Bottom view of the Electromatic proportional spacing typewriter, manufactured by International Business Machines Corp., showing the motor and power roll, cams, selector levers, escapement, etc.

CONTENTS

November, 1946 Vol. 18, Number 11

Rotary Lift Mechanism—designed for compactness 101
By John A. Draxler

Scanning the Field for Ideas 105

Engineering Record System—its design and function 108
By Russell I. Haywood

Planetary Transmissions—a design analysis 115
By L. A. Graham

Full-Flow Filtering—protects parts against wear 120

Mild Steel—its resistance to shock failure 123
By H. Markus and D. F. Armiento

Applying Quality Control Effectively 127
By T. D. Foy

Production Processes—Part XVII—Hot Upsetting 131
By Roger W. Bolz

Mechanical Seals—Part III—Application Factors 138
By Douglas R. Lewis

Fractional Horsepower Motors—new NEMA standards 143

Engineering Is Best Insurance (Editorial) 145

Design of Fluid-Film Bearings for Optimum Loading—Part II (Data Sheet) .. 151
By H. A. S. Howarth

Hastelloy Alloys—Wrought and Cast (Materials Work Sheet) 155

Itemized Index 7

Topics 98

Designs of the Month 146

Assets to a Bookcase 150

Noteworthy Patents 159

New Parts, Materials and Equipment 161

Engineering Department Equipment 174

Men of Machines 176

Business and Sales Briefs 184

Calendar of Meetings 190

New Machines 192

Helpful Literature 255

Subscription in United States and possessions, Canada, Cuba, Mexico, Central and South America: One year, $10. Single copies, $1.00. Other Countries One year, $15.

Itemized Index

Classified for convenience when studying specific design problems

Design and Calculations

Automatic stoker transmission, design of, Edit. 147
Bearings, journal, design of, Edit. 151-154
Electric motors, standards, Edit. 143-144
Filtering, full-flow, Edit. 120-122
Hot upsetting, Edit. 131-137
Mechanical seals, application factors, Edit. 138-142, 182
Mechanism, compact rotary lift, Edit. 101-104
Mechanism, crank-type multiple-spindle, Edit. 105-106
Pneumatic fatigue tester, Edit. 106
Power slush pump, design features of, Edit. 146
Quality control, applying, Edit. 127-130
Slicing machine, design features of, Edit. 148-149
Steel, shock resistance of, Edit. 123-126
Transmissions, planetary, Edit. 115-119

Engineering Department

Design service, Adv. 14, 78
Equipment, Adv. 27, 217, 229, 277
Instruments, Edit. 174
Record system, Edit. 108-114
Supplies, Edit. 174; Adv. 66, 189, 194, 248, 260, 265, 272
Testing equipment, Edit. 106, 174; Adv. 179

Finishes

Synthetic, Edit. 163

Materials

Aluminum alloys, Adv. 37, 65, 200, 201
Babbitt, Adv. 254
Brass, Adv. 88
Carbides, cemented, Adv. 62
Carbon, Adv. 226
Cobalt alloys, Edit. 166
Copper alloys, Adv. 67, 68, 79
Fabrics, Adv. 64
Felt, Adv. 252
Fiber, Adv. 221
Glass, Adv. 236
Hastelloy alloys, Edit. 155
Lubricants, Adv. 259
Magnesium Alloys, Adv. 167, 194
Molybdenum alloys, Adv. 177
Nickel alloys, Adv. 41, 240
Plastics, Edit. 161, 162, 170; Adv. 34, 202, 283
Precious metals, laminated, Adv. 219
Rubber and synthetics, Adv. 4, 24, 77, 281
Steel, Edit. 123; Adv. 35, 171
Steel, stainless, Adv. 95

Parts

Bearings:
- Ball, Adv. 6, 18, 203, 237, 267, 275, 283
- Needle, Adv. 191
- Plain, 120 degrees, Edit. 151-154
- Roller, Edit. 172; Adv. 8, 33, 93, 96, 97, 185, 199, 242, 244, 252
- Sleeve, Edit. 162; Adv. 72, 165
- Spherical, Adv. 76

Bellows, Adv. 23
Belts, Edit. 107; Adv. 15, 224, 283
Brushes, Adv. 214
Bushings, Adv. 272
Carbon parts, Adv. 10, 16, 227
Castings:
- Die, Adv. 39
- Investment, Adv. 31
- Sand, Adv. 57, 196, 206

Chains:
- Roller, Adv. 28, 82, 246

Clutches, Edit. 103, 159, 160; Adv. 26, 74, 218, 258, 263, 270, 272, 277
Compressors, Adv. 70, 193
Controls (see Electric, etc.)
Counters, Adv. 187
Couplings, Edit. 170; Adv. 56
Cylinders, Adv. 212
Electric accessories, Edit. 161, 166; Adv. 268, 270, 279
Electric controls:
- Control assemblies, Edit. 164; Adv. 69, 266, 285, Back Cover
- Relays, Edit. 168, 172; Adv. 85, 211, 245
- Solenoids, Adv. 92, 100
- Switches, Edit. 160, 162, 170; Adv. 43, 91
- Thermostats, Edit. 161, 172
- Timers, Edit. 163, 166; Adv. 192, 276

Electric heating units, Adv. 268
Electric motors, Edit. 102, 103, 107, 143, 163; Adv. Inside front cover, 1, 22, 29, 51, 73, 81, 192, 197, 198, 244, 263, 271, 274, 280
Engines, Edit. 166; Adv. 2, 19, 260, 268, 273
Fastenings:
- Locking, Edit. 159, 161, 162; Adv. 17, 53, 100, 239, 260, 266, 270, 274
- Nuts, bolts, screws, Adv. 38, 58, 173, 184, 190, 204, 205, 213, 216, 222, 262, 277, 288

Filters, Edit. 120-122, 162; Adv. 30
Fittings, Edit. 163, 170; Adv. 61, 195, 253, 267
Forgings, Edit. 131-137; Adv. 47, 210, 238, 257
Gages, pressure, etc. (see Instruments), Adv. 49, 87
Gears, Edit. 159, 160; Adv. 25, 52, 188, 250, 259, 262, 269, 276, 279
Governors, Edit. 159
Heating units (See Electric heating units)
Hose (see Tubing)
Hydraulic and pneumatic equipment:
- Controls, Edit. 168
- Pumps, Edit. 163, 168, 172; Adv. 36, 40, 45, 89, 208, 232, 234
- Systems, Edit. 106, 164; Adv. 9, 99
- Valves, Edit. 161; Adv. 54, 84, 230, 247, 258, 269

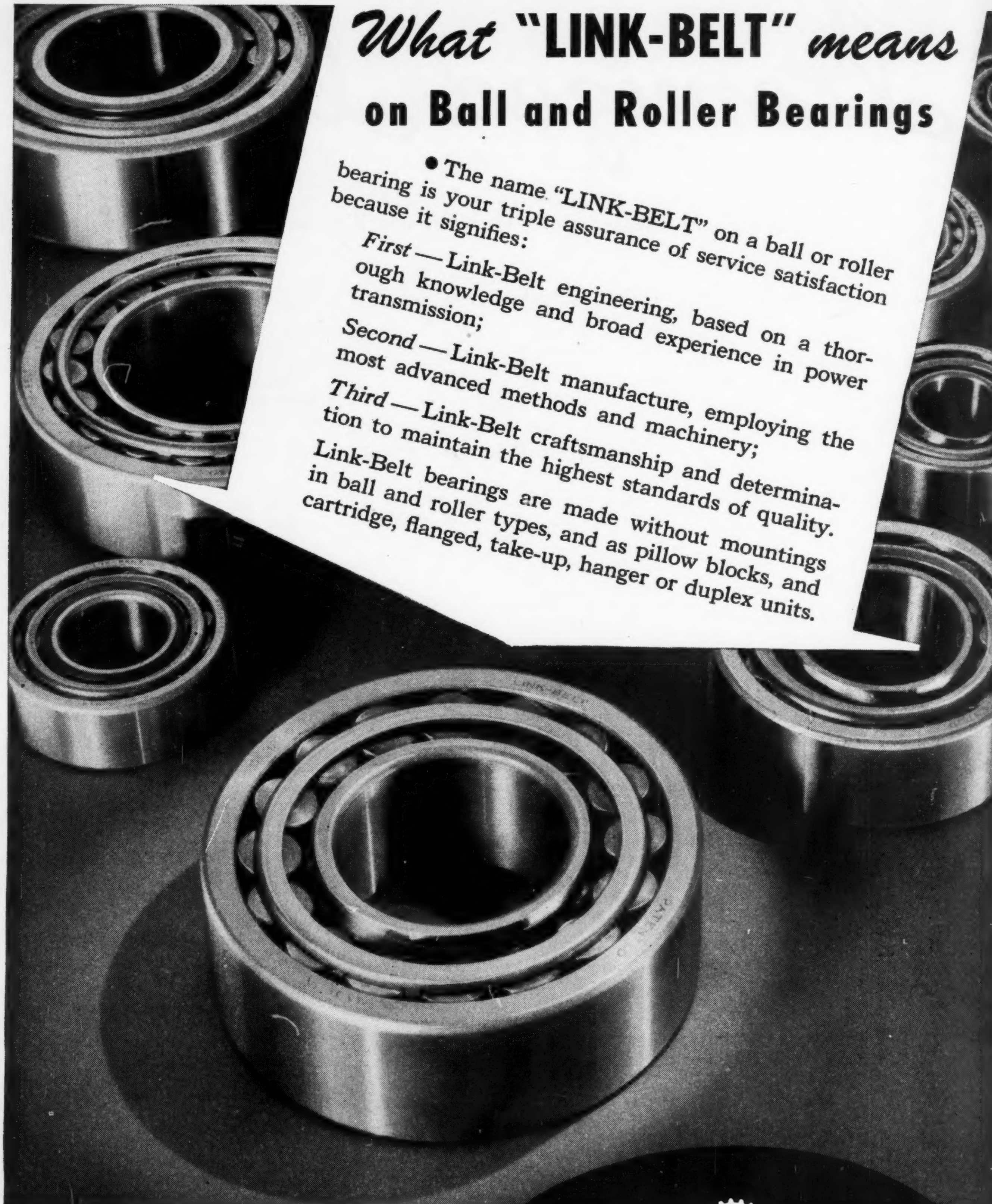
Instruments, Edit. 172
Joints, Adv. 186
Lamps and lighting, Adv. 265
Locks, Adv. 44
Lubrication and equipment, Adv. 63, 175, 263, 280
Machined parts, Adv. 215, 251, 269, 275
Motors (see Electric motors)
Mountings, rubber, Edit. 161; Adv. 21
Nameplates, Adv. 271
Plastic parts, Adv. 233
Pneumatic equipment (see Hydraulic and pneumatic)
Pumps (see also Hydraulic and pneumatic), Edit. 164, 168, 170; Adv. 262, 266, 275, 278
Seals, packings, gaskets, Edit. 138-142, 164, 182; Adv. 2, 13, 33, 169, 209, 228, 265, 271, 278
Shafts, flexible, Adv. 282
Sheet-metal parts, Adv. 273
Speed reducers, Edit. 102, 103; Adv. 11, 71, 183, 207, 225, 258, 274, 276, 287, Inside back cover
Spindles, Adv. 267
Springs, Edit. 159; Adv. 20, 250
Stampings, Adv. 46, 48, 60, 248, 280
Structures, Adv. 90
Tachometers, Edit. 168
Transmissions, variable speed, Edit. 107, 115-119, 159, 170; Adv. 5, 42, 59, 231
Tubing:
- Metallic, Adv. 55, 86
- Metallic, flexible, Adv. 235

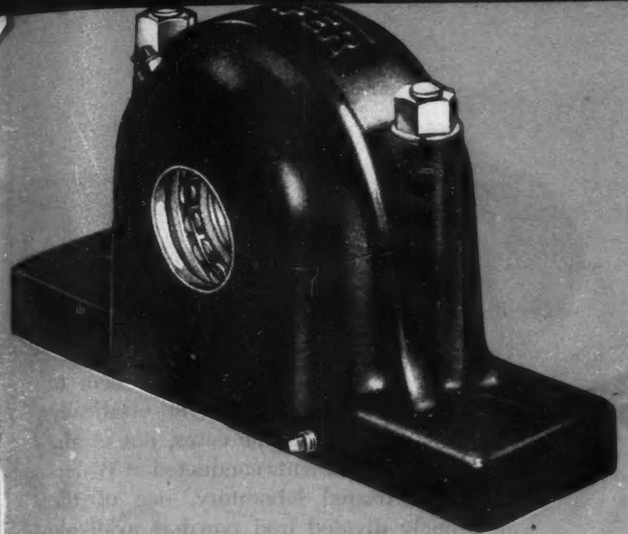
Valves (see also Hydraulic and pneumatic), Edit 162, 164; Adv. 94
Weldments and equipment, Edit. 150, 180; Adv. 12, 50, 75, 80, 181, 241
Wheels and casters, Adv. 273

Production

Broaching, Adv. 261
Hot upsetting, Edit. 131-137
Machines, special, Adv. 83, 99
Tools and accessories, Adv. 223

MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.


What "LINK-BELT" means
on Ball and Roller Bearings
● The name "LINK-BELT" on a ball or roller bearing is your triple assurance of service satisfaction because it signifies:
First—Link-Belt engineering, based on a thorough knowledge and broad experience in power transmission;
Second—Link-Belt manufacture, employing the most advanced methods and machinery;
Third—Link-Belt craftsmanship and determination to maintain the highest standards of quality.
Link-Belt bearings are made without mountings in ball and roller types, and as pillow blocks, and cartridge, flanged, take-up, hanger or duplex units.



THE SHAFER IMPROVED SPLIT PILLOW BLOCKS

Make mounting easier and lose less production time . . . provide longer life with less attention. They are niversally adaptable to line shafting and machine nstallation, dimensionally interchangeable with other tandard makes. In straight bore or adapter types. Two or four bolt design. Equipped with the new hafer "Z"-housing seal that keeps lubricant in and irt out. Equipped with the famous Shafer self-conained self-aligning DE-22000 Series bearing. They have no equal.

THE SHAFER SOLID PILLOW BLOCKS

in a wide range of sizes and capacities for all classes of service and in pedestal type where additional height between base and shaft center is needed. Rugged and compact housings for the rugged and compact Shafer double row radial thrust roller bearings with integral self-alignment that simplifies designing, machining and installing. Dimensionally interchangeable with other standard makes. Two or four bolt design. Equipped with the new Shafer "Z"-housing seal that keeps lubrication in and dirt out.

HAFER "DE-22000" SERIES SELF-CONTAINED ROLLER-BEARINGS

he Shafer self-contained unit has a one-piece outer ce, spherical inner race and two roller assemblies. readjusted, ready to install on shaft or in housings, locate machine parts and carry thrust loads. eadily interchangeable with other standard bearngs. Generous load capacity, for radial, thrust or ny combination of radial/thrust loads. Integral lf-alignment. Efficient distribution of load with no nching or binding. Natural positioning rollers sure free rolling action, no end friction.

SHAFER CONCAVEX SELF-ALIGNING BEARING ASSEMBLY

The basic Shafer bearing. The fundamental Shafer idea that merges the low rolling friction of a ball and the high load capacity of a roller into a unit that has integral self-alignment, high capacity for radial or thrust loads or any combination of the two, high resistance to shock or impact, simple in construction, easy to adjust. Available in double row as shown above, or in single row, both in a wide range of sizes and capacities.

Topics

GLASS FIBER aircraft wing has been experimentally tested at Wright field, making it the first plastic aircraft wing to meet all strength test requirements. Constructed of 55 per cent glass fiber and 45 per cent resin using sandwich design of cellular cellulose acetate set between thin skins of glass reinforced laminates, the wing poses the possibility of finding a material which can withstand the rigors of supersonic flight better than metal.

FOR NONDESTRUCTIVE TESTING, two new instruments have been developed for examining high-pressure pipe lines from the outside for internal corrosion or erosion, for exploring ship hulls or tanks for thinning of walls, and testing heat exchangers and condenser tubes for detrimental conditions. The instruments are known as the Penetron and the Probolog.

GAS-TURBINE LOCOMOTIVES, designed to run on powdered coal are expected to be in operation within the next two years. The turbines will be built by Allis-Chalmers and will be package units of 3750 hp, having a single base common to all parts —compressor, generator, gearing, and regenerator.

SUBSTITUTE HEAT AND CREEP-resistant alloys were used successfully by the Germans during the war in the construction of gas turbines and jet-propulsion engines. Having satisfactory creep strengths up to 600 C, the ferritic alloys necessitated reduction in temperatures for the turbine blade and disk, resulting in development of hollow air-cooled blades.

NONFLOWING, FINELY DIVIDED METAL powders can now be made to flow by waterproofing the individual particles, broadening the field of powder metallurgy. Condensation on metal particles causes them to flow slowly or erratically and, in some cases, not at all. In experiments conducted at Watertown Arsenal laboratory, one of the most finely divided iron powders available and typical of the powders generally considered unsuitable for powder metallurgy because of poor flow rate, was treated with a vapor of methyl chlorsilanes. After treatment the zero flow rate rose to 80 per cent of that of a standard silicon carbide powder. No impairment of mechanical properties was detected in pressed and sintered bars although there was a slight rise in apparent density.

FACTORS OF DESIGN as well as of welding have an effect on the initial formation of a crack in a welded structure. Continuation of the crack depends on the notch sensitivity of the steel, a property overlooked in past specifications for ship plate, according to investigations by the Welding Research Council and National Bureau of Standards.

RUBBER IN AUTOMOBILES is employed in approximately 265 places and additional applications are expected in future cars, according to B. F. Goodrich Co.

TYPEWRITERS to relieve the drudgery of hand-formed Chinese characters have been developed by International Business Machines Corp. with the aid of a Chinese engineer. The machine contains a cylinder bearing 5400 ideographs controlled by a keyboard of 43 keys divided into four groups. Characters are arranged on the drum in three sections, according to the frequency of use.

NOVEL BICYCLE developed in London generates and stores power in a battery for use in assisting on upgrades and for radio reception.

MINIATURE RADIO transmitters and receivers will be utilized to control Lionel toy trains. Ten radio frequencies, controlled by pushbutton on the transmitter will activate correspondingly tuned equipment located in the train.

MACHINE DESIGN

Rotary Lift Mechanism

...designed for compactness and high loads, increases versatility of industrial truck

By John A. Draxler
Assistant to Director of Engineering
Elwell-Parker Electric Co.
Cleveland

STANDARD fork-type power industrial trucks are capable of at least ten different directions of movement. The body can be run forward, turned to the left or right, or backed up and turned left or right. Headframe to which the fork is attached may be tilted forward for picking up a load, or tilted backward in carrying it. And the fork may be raised or lowered in the columns of the headframe.

Building on these basic motions, modern improvements in design have been directed toward increasing the ability of a truck to manip-

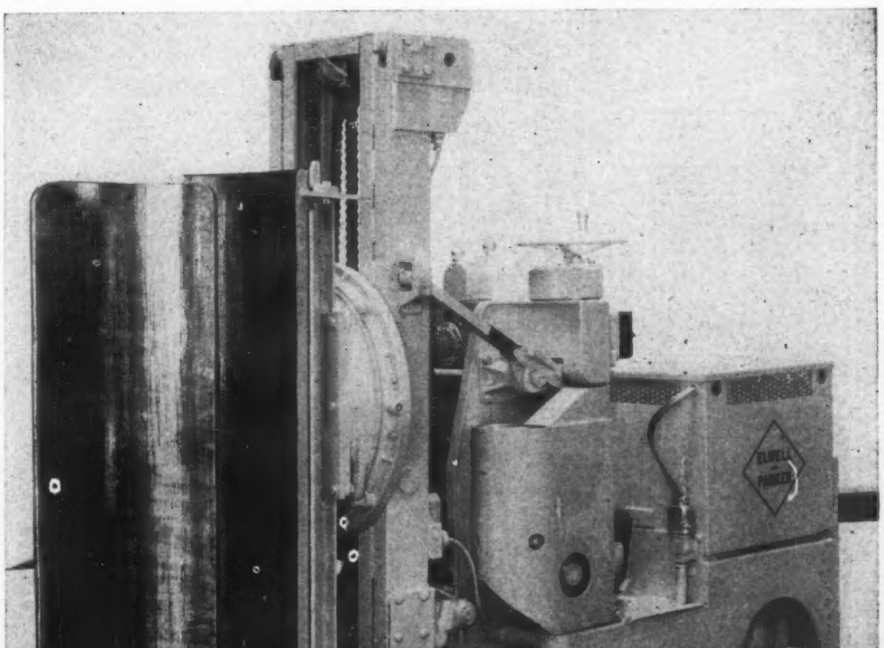


Fig. 1—Compact, self-powered rotating unit on lift adapts power truck to wide range of usefulness

ulate the more difficult types of loads. For instance, rotating a load from horizontal to vertical position or reverse, without interfering with other movements, is highly desirable when handling long and heavy cylindrical objects, such as rolls of newsprint paper and coils of strip steel. In addition, inverting steel skid boxes to discharge their contents into storage bins or road trucks often is advantageous.

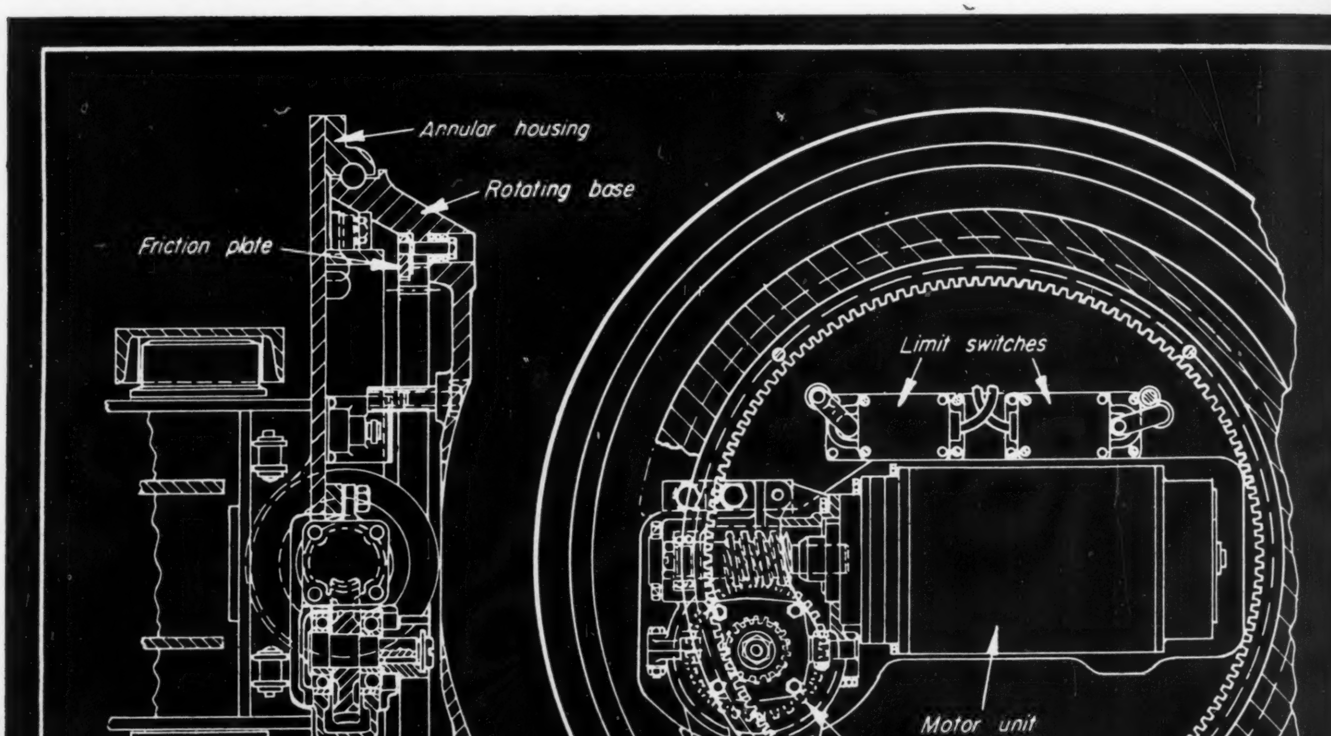
A truck which would have all these desirable features appeared to be one which utilized a power rotating unit adapted to slide in the truck's elevating frame and to which could be attached a semicircular apron for holding cylindrical objects, or a fork for engaging sleeve pallets welded to the base of a skid box. This posed the problem of developing and designing such a revolving mechanism, the most important features of which must be compactness and a capacity of 30,000 inch-pounds of torque on the rotating member.

When mounted on the front of the elevating device, the unit affects the truck's capacity for manipulating a load, reducing it because of the size and weight of the unit. It was apparent that a close-coupled member was necessary in order to retain the largest possible rating for the truck. Therefore the design of this particular feature was given considerable attention.

Another important detail of design was to make the unit, with its motor and driving mechanism, entirely self-contained and sufficiently compact to move between the upright columns of the elevating frame, *Fig.* 1. This is a distinct improvement over arrangements in which the driving motor is mounted outside the frame, in which position it interferes with the driver's view, and presents an element which might encounter an obstruction in the operation of the truck.

As designed by Elwell-Parker Electric Co. engineers, the load elevator frame has at-

Fig. 2—Cross-sectional view of revolving unit showing assembly of motorized drive inside the supporting and rotating housing



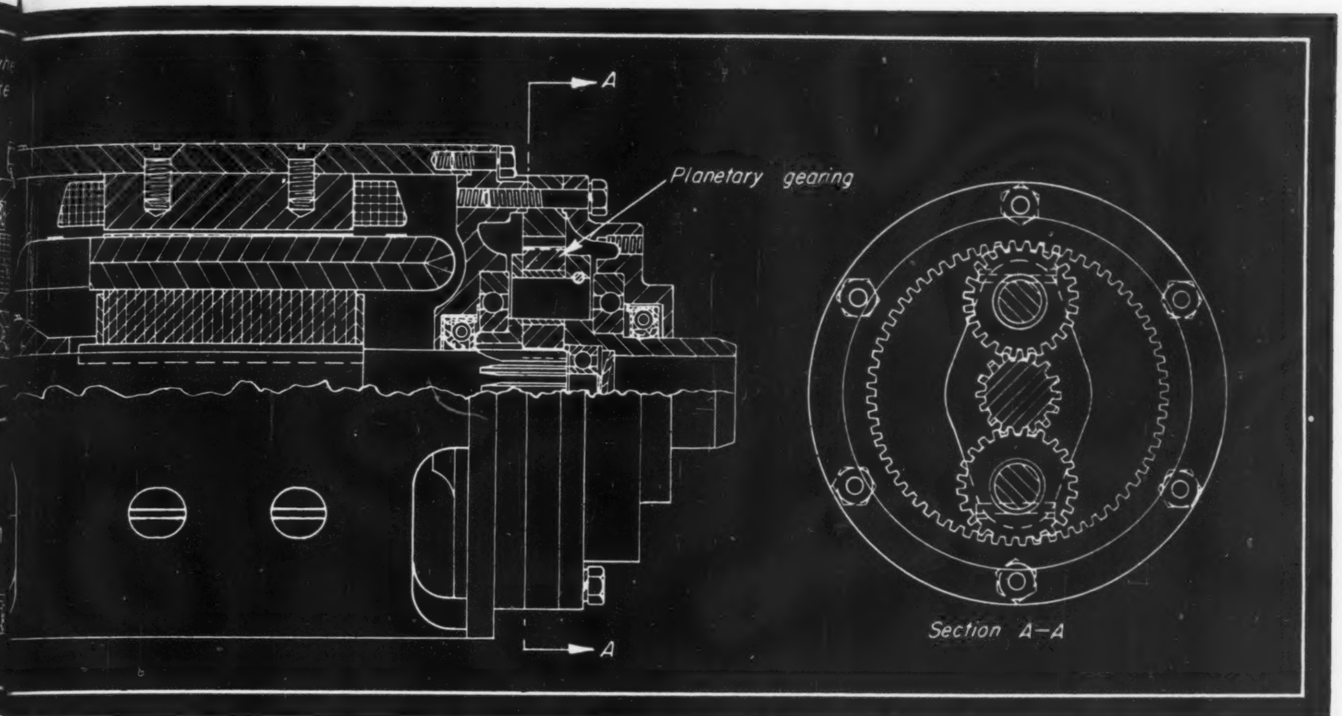


Fig. 3—Motor unit showing detail construction of the planetary reduction and magnetic brake which form an integral part of the motor

tached to it an annular housing, *Fig.* 2, which forms the outer race for a series of ¾-inch bearing balls. The inner race is a circular housing or rotating base to which is attached the apron, fork or another load-supporting device.

The particular method of journaling the rotating member on 113 balls, supported in a 27-inch diameter bearing race provides for a sturdy and stable support for the 150,000 pound-inch ultimate cantilever load supported on the apron or fork.

An electric power-operated mechanism rotates the unit about its axis. The mechanism includes an internal ring gear which is secured to the turntable through a friction holding device and a pinion for driving the ring gear. This pinion is keyed to a stub shaft, journaled in ball bearings and mounted in a housing fastened to the elevating frame.

Package Drive Is Mounted Between Channels

Also keyed to the stub shaft is a worm gear driven by a worm direct-connected to a packaged Elwell-Parker electric motor unit, *Fig.* 3. This packaged drive unit is mounted entirely within the turntable member and between the upright channels constituting the load elevating frame as illustrated in *Figs.* 4 and 5.

Operation of the motor is controlled by the operator from the platform of the truck but, to limit the rotation of the turntable automatically when desirable, limit switches are provided as shown in *Fig.* 2. These are so mounted that the current of the motor and magnetic brake will cut off when the unit has been rotated through the desired angle.

Should an overload be placed on the load-carrying member of the truck, or should it strike against a movable object, the ring gear which is held in place by a friction plate *Fig.* 2, will slip relative to the turntable housing and no damage will be done to the unit.

Package Drive Unit: The drive unit, *Fig.* 3, which is the heart of the revolving mechanism, utilizes a specially built motor with a 5:1 ratio planetary back gearing and an electric solenoid brake made integral with the motor. The planetary unit, although an integral part of the motor, operates in a bath of oil within a sealed chamber. The planetary spider is full ball bearing mounted and utilizes two 24-tooth, 16-pitch planetary pinions, having a ⅝-inch tooth face. These pinions are high-grade alloy steel, carburized and hardened for maximum strength and wear.

Located at the opposite end of the motor and incorporated within it is an electric solenoid brake to hold the armature shaft, thereby locking the rotating member in any position within its range of rotation. This brake consists of a brake wheel keyed to the end of the armature shaft and a stationary armature plate, *Fig.* 3, containing a cork brake disk which presses against the brake wheel by action of a series of springs when the brake is engaged.

When the current supply to the motor is interrupted, the electro-magnetic brake automatically holds the armature shaft in its position. When the current supply is resumed, the coil again becomes energized and withdraws the armature from its braking position against the pressure of the springs.

Fully Class B insulated, the motor is series wound of oversize bar copper for 30 to 36-volt operation with a speed of about 2000 rpm on the armature shaft or an output speed of 400 rpm obtained through the planetary reduction unit.

Worm Reduction Unit: The worm-gear reducer is also a self-contained unit containing a 29-tooth bronze worm wheel, *Fig.* 2, and a hardened alloy-steel worm. All moving parts are ball bearing supported and operate within a sealed oil chamber. The worm is direct connected to the packaged power unit through a splined fitting. When assembled the two self-contained units form a master power

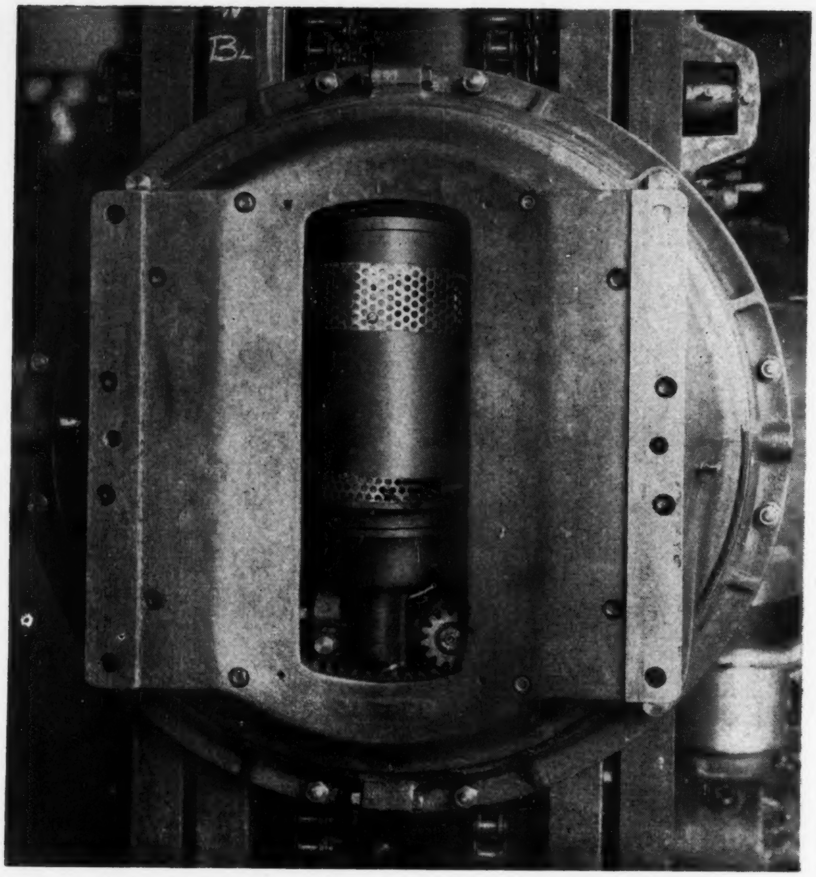
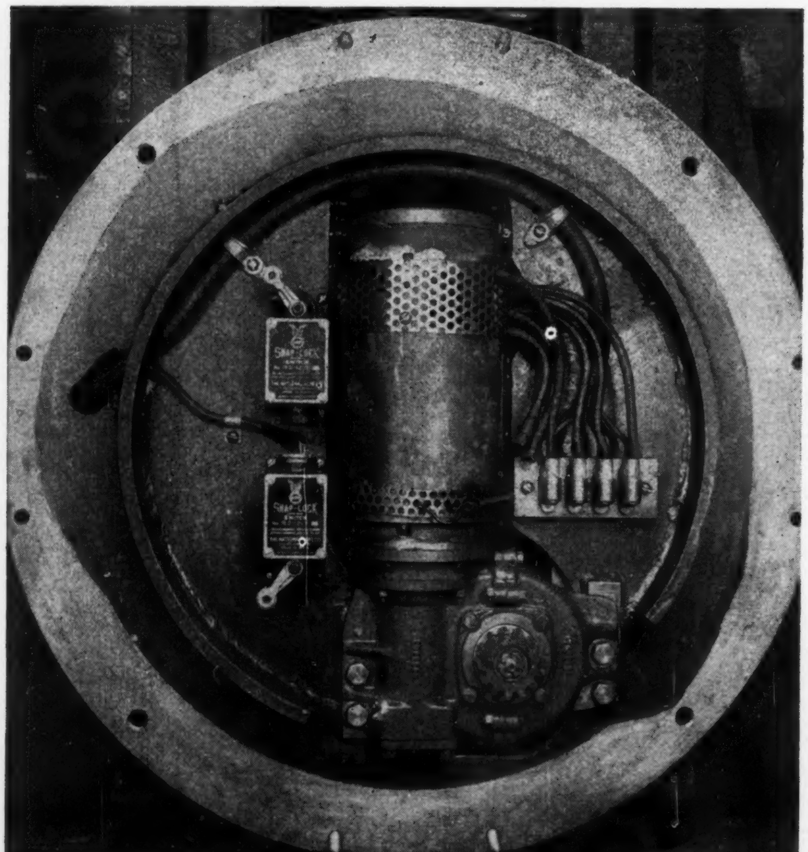


Fig. 4—Above—Assembly of electric motor and worm-gear auxiliary reduction unit inside the elevator frame

Fig. 5—Below—Motor and auxiliary worm-gear reducer in rotating unit. Rotating member and annular housing supporting the ball bearings have been removed. Limit switches control degree of rotation



unit having a total reduction of 145:1.

The rotating feature of this power truck has increased its flexibility and has enabled an operator to manipulate loads formerly considered "impossible". When equipped with roll-handling apron and scoop, *Fig.* 1, the truck can pick up, carry, store, or stack newsprint rolls 36 to 42 inches in diameter, 6 feet or more in length and 2000 pounds in weight. It can lift such a load in either horizontal or vertical position to a height of 107 inches. With apron removed and fork attached to the rotating member, the truck has capacity for lifting 3000-pound loads.

No sooner did this rotating device appear than it was applied to handling bulk loads in steel skid boxes in novel manner. An operator can transport, elevate and turn the boxes upside down to empty the contents, as the built-in sleeve pallets prevent the boxes from falling off the fork. Numerous other appliances also have been developed to utilize this new rotary feature.

Seeing Against the Sun

ENABLING PILOTS to spot enemy aircraft, speeding toward them directly in line with the sun, a new device called the Icaroscope utilizes a transparent phosphor screen alternately exposed to the image and the pilot's eyes. This instrument, described by Dr. Brian O'Brien at the annual meeting of the Optical Society of America, employs a double rotating shutter driven by an electric motor at 100 cycles per second. Image exposed during one phase of the shutter is visible one-hundredth second later by residual phosphorescence. The brilliance of the sunlight on the phosphor screen is only 20 to 50 times that of the surrounding sky, compared to the sun's real brightness which is 10,000 to 100,000 times that of the surrounding sky. The result is that a plane can be seen on the screen silhouetted against the sun's disk or the surrounding sky.

Viewing appears to be continuous as in a high-speed motion picture projection. Operation resembles that of the classical polariscope with the important difference that the entire image of a scene is formed upon a stationary screen and the phosphor is chosen to have short afterglow when saturated.

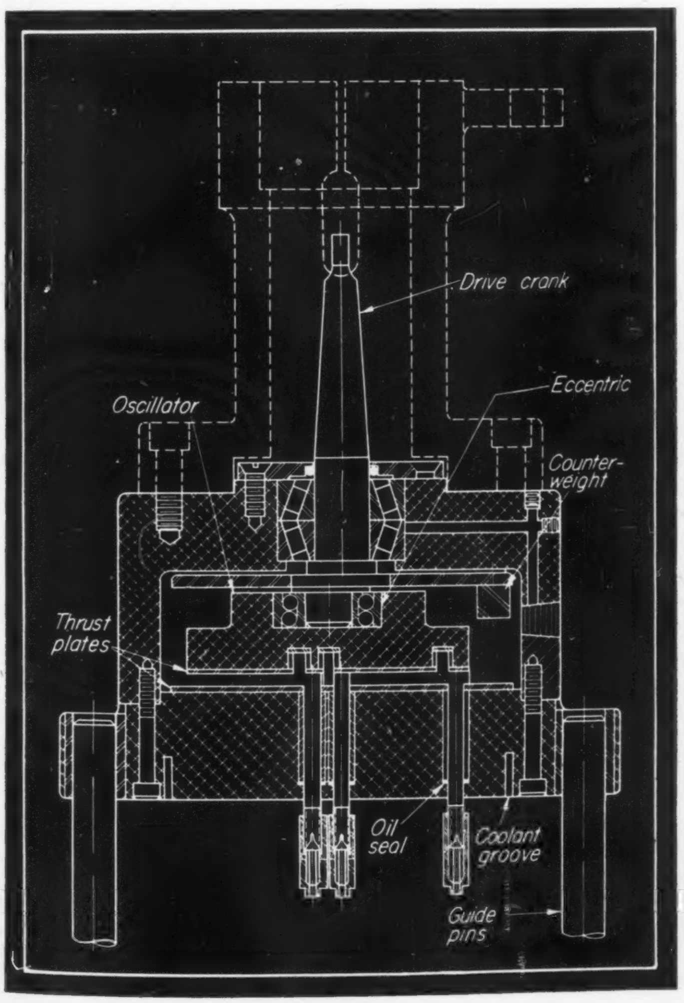
The Icaroscope was developed at the University of Rochester's Institute of Optics under a contract with the Office of Research and Development as an aid in defense against aircraft attacking from general direction of the sun. It was used extensively in observing and photographing the Bikini atom bomb tests.

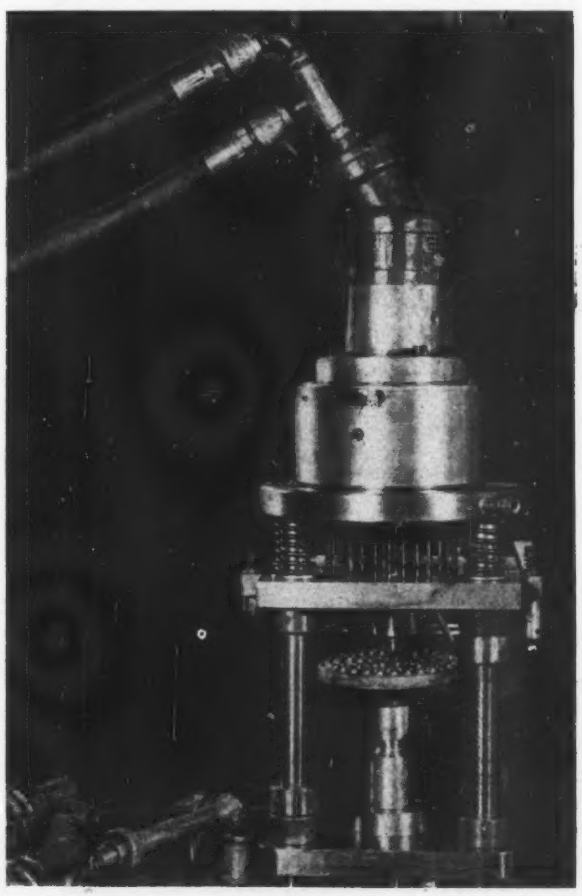
Scanning THE FIELD for Ideas

Multiple-spindle drilling heads using a crank type drive are not new in principle but for some reason or other commercial designs heretofore have never been developed. Generally, mechanical construction was poor for the individual units that were built. Some designs utilized drills with their shanks bent to form a crank but, because they offered no bearing thrust, were subject to excessive maintenance.

In the drilling head illustrated below, Zagar Tool Inc. has overcome these apparent weaknesses and has standardized design to take care of 80 per cent of multiple-spindle drilling requirements. First consideration in design was the spindle bearings themselves. Center distances are based on standard needle bearings, giving a minimum spacing of 3/8-inch. For closer spacing, bronze bushings are utilized, allowing 1/8-inch drills to be set on 1/4-inch centers.

By providing a surface on the arm portion of the crank, thrust is absorbed and adequate





bearing surfaces are obtained both on the drill crank and on the bronze bearing thrust plate. Action of the thrust surface is one of generating itself about the axis of the crank pin. Resultant motion is a circle which is approximately three times the area of the thrust surface.

Vibration of the head is reduced to a minimum by the use of a counterweight on a balance plate. Power is transmitted through the drive shaft which oscillates the driving plate in a small circular pattern to revolve the drill spindles. Guide pins and bushings are employed to save setup time and reduce wear on the drill jig and the drill head. Because there are a larger number of spindles working at close centers in a comparatively small space, a water cooling jacket is provided to dissipate the heat.

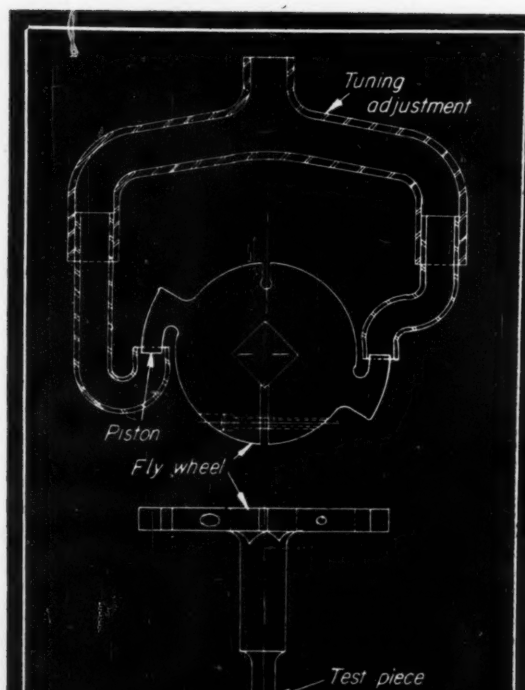
Tuned air columns in pneumatic circuits have been found to simplify the design of vibrating and torsional fatigue testing machines. The principles employed in these two types of machines are shown schematically below. Developed by the General Electric Co. for testing turbosupercharger buckets, etc., these pneumatic machines have such efficiency that testing time is materially reduced.

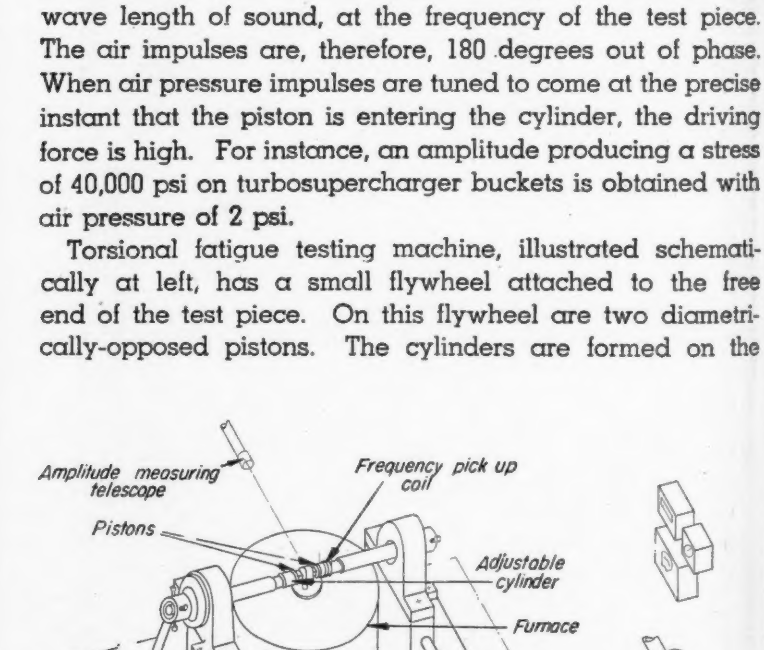
Tuning is accomplished by decreasing or increasing the length of the air path much as a trombone player changes the tone of his instrument by moving the slide. Part to be tested is placed in position between two air paths. It vibrates or oscillates until failure occurs, at which point the natural frequency of the piece changes and is out of phase with the air column. The piece then ceases to vibrate.

The vibration testing machine, shown below, has the cylinders or air paths mounted on ball seats so that they may be easily aligned. Pistons are attached to the free end of the test piece. On the end of each cylinder is an adjustable sleeve, allowing the operator to back off the cylinder as the amplitude is increased. To measure the frequency a small coil of wire and a magnet is mounted on one of the sleeves. When failure occurs, manifold pressure increases because of the decreasing frequency. A mercury manometer responds to this pressure rise and closes a relay contact, shutting down the machine.

Opposed cylinders are used to make the machine stable and to give an impulse every half cycle. As shown, two cylinders are connected to a common manifold, receiving air at a constant pressure from a pressure-control valve. The total air path from one cylinder mouth through the tuning device to the other cylinder mouth is equal to one-half the wave length of sound, at the frequency of the test piece. The air impulses are, therefore, 180 degrees out of phase. When air pressure impulses are tuned to come at the precise instant that the piston is entering the cylinder, the driving force is high. For instance, an amplitude producing a stress of 40,000 psi on turbosupercharger buckets is obtained with air pressure of 2 psi.

Torsional fatigue testing machine, illustrated schematically at left, has a small flywheel attached to the free end of the test piece. On this flywheel are two diametrically-opposed pistons. The cylinders are formed on the

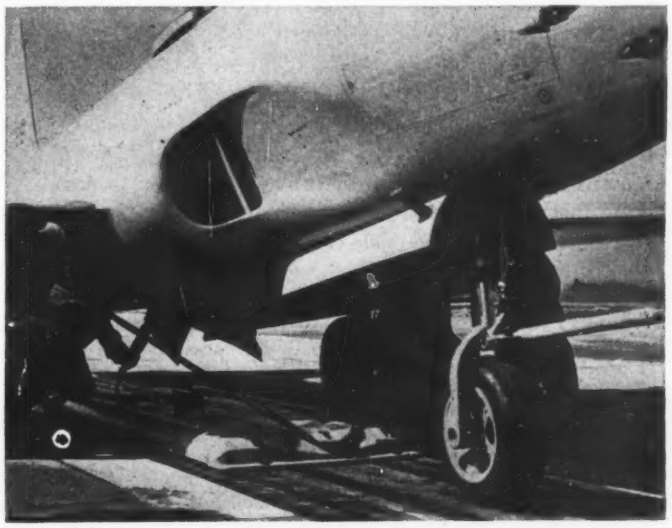




same arc as the pistons to maintain constant clearance. Operation of the machine is the same except that the air impulses are in phase. It is important to keep the air impulses in phase, otherwise a bending moment would be superimposed on the torsion.

Change of air path length is necessary because the natural frequency of the test piece drops with rise in temperature. For bucket test pieces the frequency drops form 250 to 210 cycles per second between ambient temperature and 1350 F.

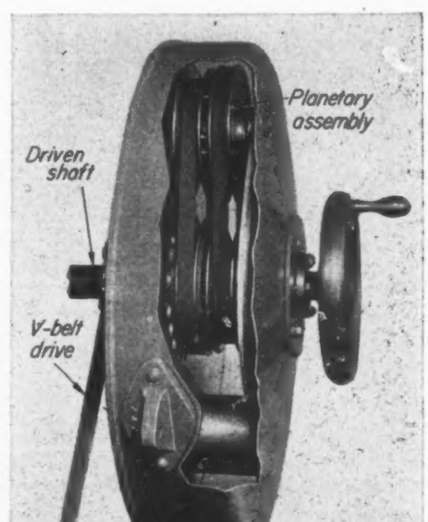
Linear electric motor, 1382 feet long, launches aircraft from small fields without the initial impact of conventional catapults. Called the Electropult and designed by Westinghouse Electric Corp. for the Navy, the device corresponds to a huge motor laid out flat. In operation, a shuttle car which acts as a stator is hitched to a plane by a bridle as shown at right. Jet-propelled planes have been launched at 116 miles an hour after a run of 340 feet. Running without load the shuttle car develops a speed of 226 miles an hour in 500 feet.



Installed at Patuxent River Base, the track is mounted flush with the concrete runway and above a trench carrying copper bus bars for the motor. On each side of the track and beneath its surface are rails to carry the car. The car itself is $11\frac{1}{2}$ feet long and extends only five inches above the track. When launching, the bridle drops off the plane after flying speed is reached.

Power supply for the motor is obtained from a Pratt & Whitney 1100 hp aircraft engine driving a direct-current generator, connected to a motor. This motor in turn drives an alternating-current generator and a 24-ton flywheel. The flywheel stores sufficient kinetic energy, when accelerated to 1300 rpm, for the generator to supply 12,000 kw to the Electropult during the few seconds of plane launching.

Planetary V-belt, variable-speed reduction unit illustrated at right gives stepless speed changes from full forward through zero to full reverse at constant torque of two horsepower when driven at 1100 rpm. Two belts and four variable-pitch pulleys are utilized in this planet system developed by Speed Selector Inc. The unit is mounted on the driven shaft and drive is from motor through a V-belt to a pulley and quill assembly. Slight change in the variable-pitch pulleys, obtained by adjusting the handwheel produces a large change in output speed through the action of the planetary. Difference in the speed of the pair of belts is applied to the output shaft. When the belt speeds are equal the output shaft speed is zero. Handwheel, mounted on the input shaft of the driven machine by a tapered collet, alters the pitch diameters of the center pulleys so that as one is increased the other is decreased. Change is imparted to the outer pulleys by the wedging action of the two belts. Thus no complex linkages or springs are needed. Housing of the unit supports the hand wheel and is held stationary by a bracket mounting. Outer pulleys are counterbalanced by a weight shown in lower portion of the cutaway view below.



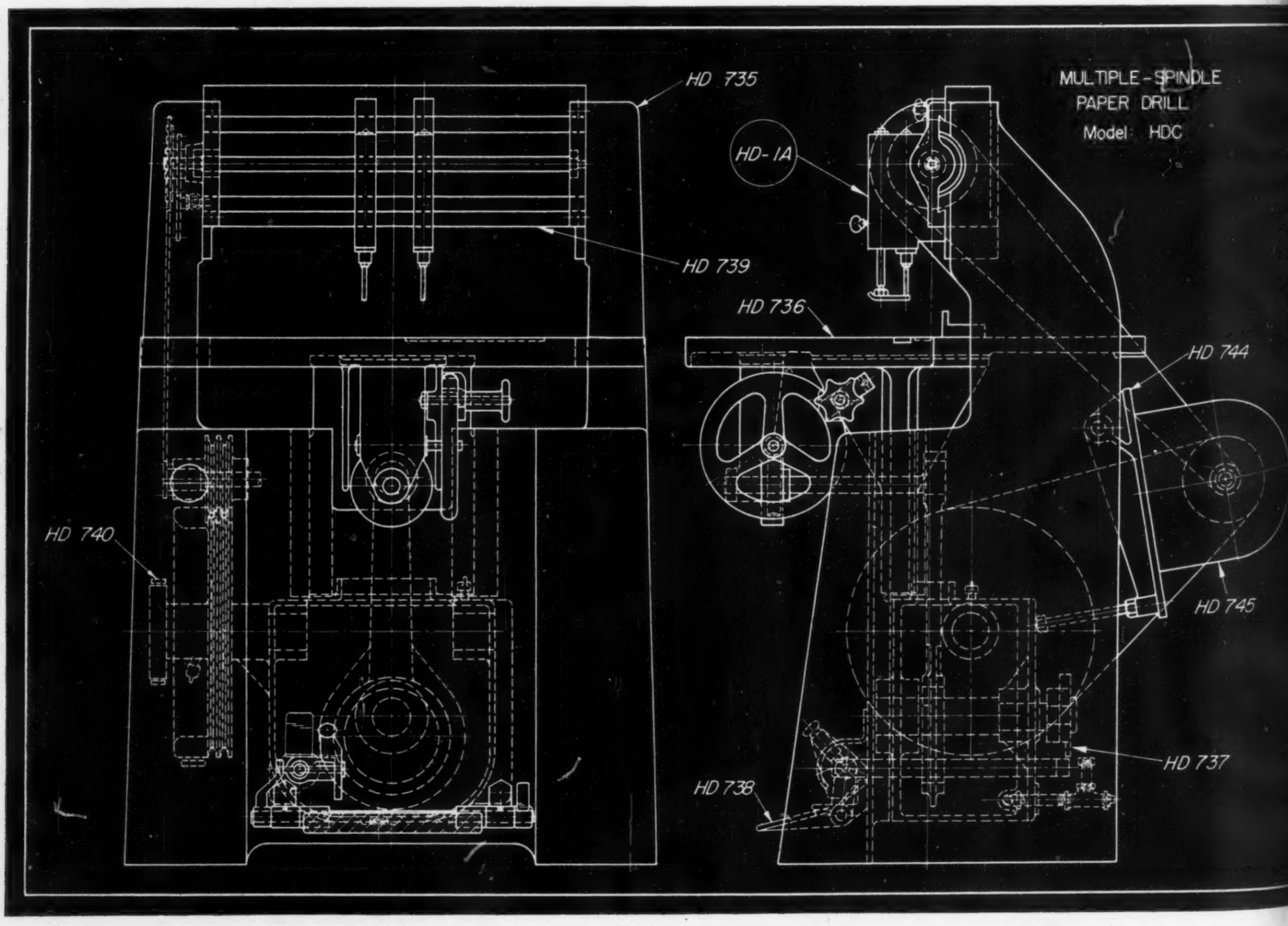
Engineering Recor

By Russell I. Haywood
Assistant Chief Engineer
Seybold Div., Harris-Seybold Co.,
Dayton, Ohio

HIT-OR-MISS engineering record systems, with much of their reference data filed in the memories of long-time employees, may ineptly suffice for the small machine manufacturer. For the larger plant, however, only a competently designed record system will meet the needs of modern competitive industry. The system herein described, while presented particularly as a guide for those plants which have mushroomed during the war, is also offered as a comparator on contemporary practice for all manufacturers

ENGINEERING records—that accumulation of knowledge covering the manufacture of products over a long period of years—are the life blood of today's leading manufacturing plants. Engineering records substantiate the progressive company's philosophy that it must perpetuate itself and must not be restricted to the lifetime cycle of any individual or group

Fig. 1—Below—Drawing of model identifies all units. Their symbol is first two letters of the model symbol



System

of individuals within the organization.

An engineering record system must be economically sound and flexible to the extent that it can be used for the very small as well as the largest operations. Engineers have made drawings and kept records of them in some form or other for scores of years without realizing the potential value of a modern system. Engineering records are made up of a complex, diversified array of drawings and other forms containing such important information as machine indexes, bills of material, lists of part numbers, optional equipment lists, engineering releases, change notices, and other valuable data.

In establishing the system for compiling and maintaining records or engineering drawings which is described in this article, the major objectives were to provide a system that would be simple, direct, complete, and of such form that all departments could use it without change. Many other benefits are derived from this system, such as:

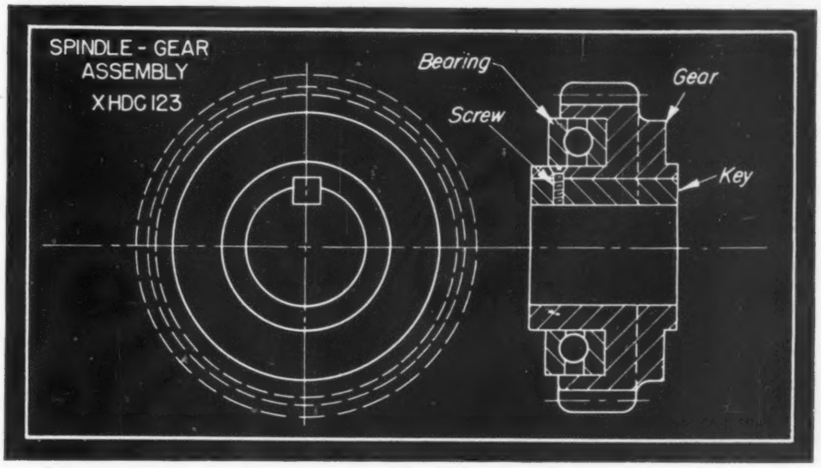
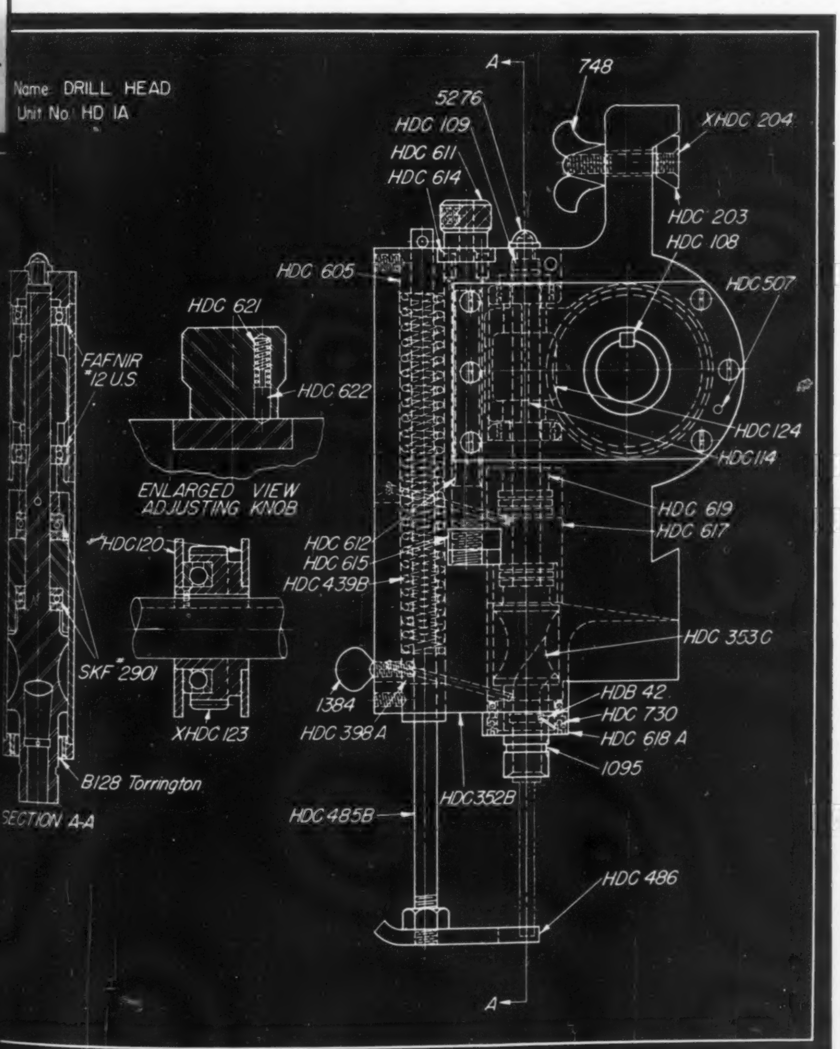


Fig. 3—Above—Drawing of assembly used in unit HD-1A (see Fig. 2). Number of assembly is that of its major part, the spindle gear, preceded by the letter X

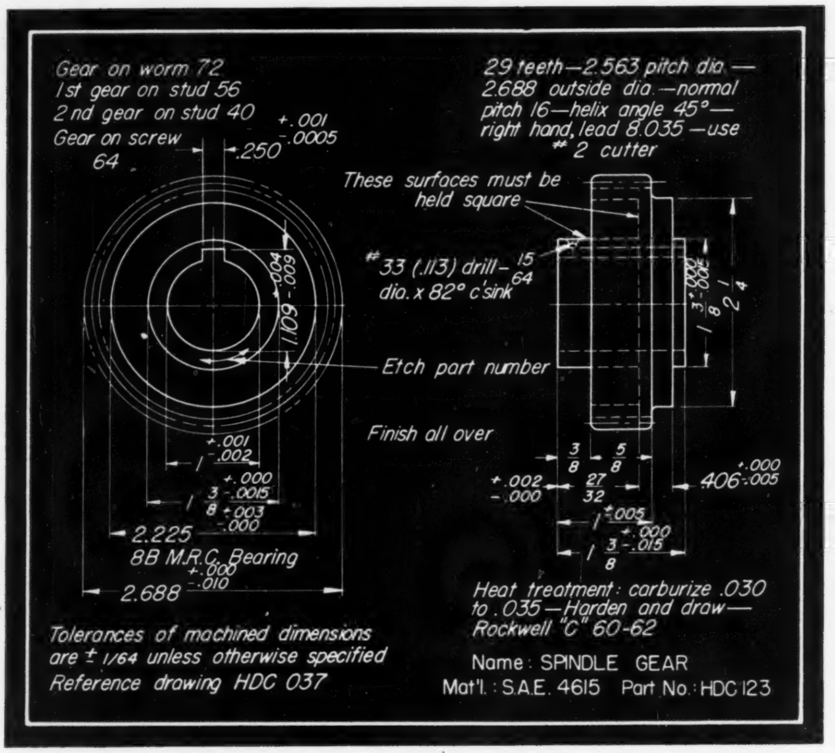
Fig. 2—Below—Unit drawing (number circled in Fig. 1), identifies all assemblies and parts which are used in unit



1. Elimination of the possibility of errors, made when information must be converted into other forms, schedules, order requisitions, etc.
2. Elimination of extra clerical work in setting up duplicate information
3. Elimination of excessive loss of time by the engineering department in answering routine questions
4. Elimination of the need for technical "know how" in departments other than engineering
5. Avoidance of delays in procuring materials vital to production (especially important in long-cycle manufacturing operations)
6. Provision for accurate cost and pricing records by definitely establishing the product.

Special consideration was given to records pertaining to parts commonly referred to as "hardware shelf" items. It was felt that the relatively low cost of these items would obviate the necessity for cataloging information regarding them. However, it was also realized that a sudden exhaustion of the "hardware shelf" supply, that could not be replenished in time to meet production schedules, would be comparable to a shortage of a major part.

First step in setting up the record sys-



tem was to divide each product and its drawings into four basic elements for record purposes: Models, units, assemblies, and parts.

Model: A complete machine. Its drawing carries three letters for identification. The multiple spindle paper drill shown in *Fig.* 1, for example, carries the letters HDC.

Unit: A major grouping of parts within a model. It comprises either: (A) A major assembly, including installation parts, or (B) assemblies and associated unassembled parts, which function as a group on a specific model or are common to machines of a type. The unit symbol

Fig. 4—Left—Part drawing of spindle gear used in assembly XHDC-123 (Fig. 3) takes its symbol (HDC) from the model

Fig. 5—Below—Model-index bill of material (B/M) lists only the units which comprise a complete model. This one, carrying the number HDC, applies to the multiple-spindle paper drill of Fig. 1

HARRIS • SEYBOLD CO. DAYTON, OHIO — **BILL OF MATERIAL** — LOT — S.O. NO. — DATE FILL'D — QUAN.

LINE	PART NO.	QTY PER UNIT	A B C D NAME	(PATT. NO.)	MAT.	TOTAL QUAN.	ORDER	OK	UNIT COST	MAT'L	BURDEN
1	HD 735	1	Frame								
2	HD 736	1	Table								
3	HD 737	1	Gear box								
4	HD 738	1	Treadle								
5	HD 739	1	Chip conveyor								
6	HD 740	1	Brake								
7	HD 741	1	Back gage								
8	HD 742	1	Table illumination								
9	HD 1A	2	Drill head								
10	HD 744	1	Motor bracket								
11											
12			SELECT ONE PER ORDER								
13			Drive Motor Equipment								
14	HD 745	1	220/440 volt 60 cycle 3 phase								
15	HD 746	1	220/440 volt 50 cycle 3 phase								
16	HD 747	1	220/440 volt 25 cycle 3 phase								
17	HD 748	1	550 volt 60 cycle 3 phase								
18	HD 749	1	550 volt 50 cycle 3 phase								
19	HD 750	1	550 volt 25 cycle 3 phase								
20	HD 751	1	230 volt D.C.								
21	HD 752	1	208 volt 60 cycle 3 phase								
22	HD 753	1	220/440 volt 60 cycle 2 phase								
23											
24											

PREP. BY J.C.	SUPER	NAME MULTIPLE SPINDLE PAPER DRILL	
TYP. BY M.S.	ISSUED	DWG. HDC 18x24	SHEET 1 OF 1 — NO. HDC
POSTED BY			
CHK. BY	SUP. BY		

FORM NO. F-175-A

and number is the first two letters of the first model on which the unit is used, followed by a number which has no special significance. The drill head, shown in *Fig.* 2, is a unit of the multiple spindle paper drill (HDC), therefore carries the symbol HD.

ASSEMBLY: A group of preassembled parts that function together within a unit. Assemblies are created only when: (A) Manufacturing operations after assembly are required, (B) manufacturing or stocking the assembly is desirable either for subsequent erection or service, and (C) parts are joined permanently by welding, brazing, etc. The symbol and number of an assembly is the part number of its major part preceded by X. If the same major part is used in more than one assembly, however, another number is chosen, still preceded by X. The number then has no special significance. *Fig.* 3 is an assembly within Unit HD-1A, and its major part is the spindle gear HDC-123 (*Fig.* 4). Thus the assembly takes the number XHDC-123.

PARTS: These, of course, are the individual pieces required to make up an assembly or a unit. Drawings always are made of the individual pieces. Each drawing, *Fig.* 4, is assigned a part number which is composed of the symbol (3 letters) of the first model on which the part is used, followed by a number having no special significance. When the design of a part is changed and a record of the change must be kept, a new drawing is made and a suffix added to the original part number to form the new part number. Obviously, in no case is it permissible to assign the same number to more than one part, or more than one number to the same part.

Fundamental components of the Harris-Seybold drawing record system are:

1. Bills of Material
2. Drawing Record Cards
3. Change Notices
4. Engineering Releases.

Bills of Material Have Maximum Utility

BILLS OF MATERIAL: These are compiled and maintained by the engineering department. They are the basic and only authentic records which list the parts making up our machines and show their general arrangement. Since these forms are used by all departments, they are designed for maximum general utility. Some of the departmental uses are:

Engineering department: Machine specifications, optional and standard equipment information
Factory: Manufacturing, procurement, tooling, assembly, erection, and standards
Cost department: Cost of machines, units, assemblies, parts, and pricing
Service department: Permanent machine records
Branch Offices: Service and installation information.

Three kinds of B/M (Bills of Material) are maintained:

1. Model Index B/M.
2. Unit B/M
3. Assembly B/M

Model Index B/M's list only the units which comprise a complete model. That shown in *Fig.* 5, carrying No. HDC,

NUMBER	DWG. NO.	SIZE	PATTERN
HDC 123		9 x 12	

NAME
SPINDLE GEAR

SUPERSEDED BY	SUPERSEDES	1 CARD OF 1

DATE	USED ON	Q	DATE	USED ON	QUANTITY
3-7-45	XHDC 123	1			
3-7-45	HD 13 A	2			
1-10-46	HD 27 A	3			

Fig. 6a—Above—Front side of Drawing Record Card carries pertinent data such as identification number, drawing size, etc. Fig. 6b, below, shows reverse side of card which lists microfilm record

MICROFILM RECORD OF				
REV.	ROLL	PLACE	DATE	REMARKS

ADDITIONAL INFORMATION

H. S. P. FORM 481

applies to the multiple-spindle paper drill shown in *Fig.* 1. Unit B/M's list all parts required for an assembly and also list subassemblies with their parts. Separate assembly B/M's are written only when the assembly is not shown on a unit B/M. This occurs when: (a) Special assemblies are required for service, (b) assemblies are made of optional equipment, and (c) an assembly number is required before the assembly grouping is made on the unit B/M, as a temporary measure.

Taking the various entries in order, as shown in *Fig.* 5, the arrangement of a B/M is as follows:

Part No.: This, entered in the left-hand column, is the Harris-Seybold part number as used by this company. Vendors' part numbers are listed in the name column.

Qty. Per Unit: The quantity shown in this column is the total number used in the Unit, in this particular grouping of parts. This rule applies whether a part is separate or within an assembly.

Name: The name of a part or an assembly should correspond to the name shown on its drawing. Standard parts may incorporate usage description in the name even though this does not appear on the drawing. The identifying noun usually should appear first. In the case of an assembly, the word "assembly" should follow the name.

Indention: Components of an assembly are identified by indenting or offsetting their names one column to the right of the assembly name. Components of subassemblies within other subassemblies are similarly identified by indention until all individual parts are listed on the B/M. This shows, then, in one indention column, the names of

FORM NO. CP. 205 HARRIS SEYBOLD COMPANY SEYBOLD DIVISION DAYTON, OHIO		COPIES TO	B/M	B/P		COPIES TO	B/M	B/P		COPIES TO	B/M	B/P
	X	COST			X	PLANNING		X	X	SUPERTND'T		
	X	ENGINEERING			X	POC		X	X	TOOL DESIGN	X	X
	X	ERECTION			X	PROCESSING		X		TOOL ROOM		
	X	FOUNDRY			X	PURCHASING						
	X	INSPECTION		X	X	SERVICE	X					

DATE ISSUED 9-23-46

ENGINEERING CHANGE NOTICE

	DESIGN OR SPECIFICATION CHANGE & REMARKS	NEW PART NO. OR B/M NO.	CHANGE	OLD PART NO. OR B/M NO.	QUANTITY	EFFECTIVE	NAME AND REMARKS	B/M REF. PAGE	B/M REF. LINE	DISPOS. PARTS IN STOCK	PARTS IN PROCESS	IN STOCK
1	Material Changed and		P	HDC-124	/	A	Spindle Gear	1		D	D	-
2	Notes added.											
3												
4	Thread Length		P	W-1384	/	B	Thumb Screw	1		C	C	-
5	changed 3/4 to 7/8											
6												
7	Material Changed	HDC-120A	V	HDC-120	2	A	Side Plate	1		H	H	
8	From Steel to Brass											
9												
10												
11												
12												

* NOTE

MODELS AFFECTED HDC Drill

LEGEND

*	SEE NOTE	E	MAKE TO REVISED SPECIFICATIONS
✱	ADD		
—	REMOVE	F	USE FOR SERVICE
V	SUPERSEDES (NOT INTERCHANGEABLE)	G	NEW PATTERN NEEDED
		H	SCRAP
O	REPLACES (INTERCHANGEABLE)	K	S.O.S. NO.
		M	ON LOT
A	AT ONCE		
B	ON NEXT SHOP ORDER FOR PART	P	PRINT REVISION MACHINE NO.
C	USE AS IS		
D	REWORK TO NEW PRINT		

REMARKS & REASON FOR CHANGE: ENG. WORK ORDER NO. 186

LINE NO.	
1	Heat Treat note added — Eliminate wear and
	Co. Sink Enlarged to Sink Screw Head.
4	More Adjustment.
7	Improve Bearing.—Send New Plates on Service.

REQUESTED BY	CHG. REQ. NO.	WRITTEN BY	APPROVED BY	SHT. 1 OF 1	ECN NO.
R. I. H.		J. L.	W. W. R.		W - 92

Fig. 7—Engineering change notice is used to inform all departments of changes and revisions applying to records as well as drawings, thus has maximum utility

all the parts that go to make up an assembly. Insofar as possible, units and assemblies and their components should be listed in the B/M in a sequence determined by their relationship to each other.

Attaching or installation parts of an assembly are shown on the same indention column as the name of the assembly. If a unit consists of a single assembly plus its installation parts, the assembly is given a number and the installation parts are on the same indention column as the assembly name. Blank lines between parts names should have no special meaning and can be used to provide space for additions or corrections. Alternative parts or parts associated with different sizes or models of machines may be shown, provided they are enclosed by brackets and accompanied by a note such as "Use one only as per order".

Patterns: When a part listed is made from a casting of any type, the pattern number, enclosed in brackets, should be entered after the name, if it differs from the part number.

Materials: If a part is made from a casting or a forging, this is indicated by posting the letter C or F in this column. The next eight columns are left blank for P.O.C. (Production Order Control) and Cost Department entries.

Revisions: As changes or revisions are made on a sheet of a B/M, a code number is posted in the revision column opposite the part revised. Subsequent changes are numbered in numerical order, each sheet having its own series. The number is also posted in the center lower block, starting in the space under the word "issued". The Engineering Change Notice number is posted in the next column to the right, and the date of posting the change is entered in the third column.

Name of Unit: This is shown in the lower right-hand block. It is the same as that on the unit drawing if one has been made. If there is no drawing, the note N/D should be entered in the "Dwg." block. If the drawing number differs from that of the unit, it, as well as the size, should be noted in the "Dwg." block. Numbering of the unit is handled exactly as a part number with suffixes for changes, etc.

Drawing Record Cards: Drawing Record Cards have the following purposes:

(a) Assigning of numbers to parts, assemblies, and units
(b) Listing of drawing size to determine file location
(c) Listing of pattern information
(d) Listing of information regarding status of item
(e) Listing of all records showing the usage of the item

(f) Listing of the microfilm record
(g) Listing of additional and special information.

As is illustrated in *Fig. 6a*, the number assigned to an item is entered in the "number" location on the Drawing Record Card. The record of all numerical assignments is maintained on these cards only. New parts, assemblies, or units take the next number in their sequence showing in the card file. Changed parts carrying a suffix letter constitute a part number change and are treated as new parts by making new record cards. Cards may be removed from the file only for making entries or if a substitute card replaces it temporarily.

The "Drawing No." is entered only if it differs from the item number. Drawing size is shown in the "Size" location as 9 x 12, 12 x 18, 18 x 24, etc. If there is no drawing, as on some assemblies or units, N/D is entered in this box.

Pattern number is entered only if it differs from the part number. This may happen when a part number has been changed, but not the pattern, or when one pattern is used to make castings for two or more parts. In the case of assemblies for which there is no separate B/M nor usage in higher records, the note "No B/M" is entered in the space for the size.

When a new item supersedes or replaces an old item, the old item number is entered in the "Supersedes" location on the new card and the new item number is entered in the "Superseded by" location on the old card. When a new item replaces an old item, that is, when they are interchangeable in all usages, the word "Dead" is stamped across the face of the card for the old item. When more than one card is needed to carry the record of any one item the different cards are identified by entering 1 of 2, 2 of 2, etc. in the space marked "Card of".

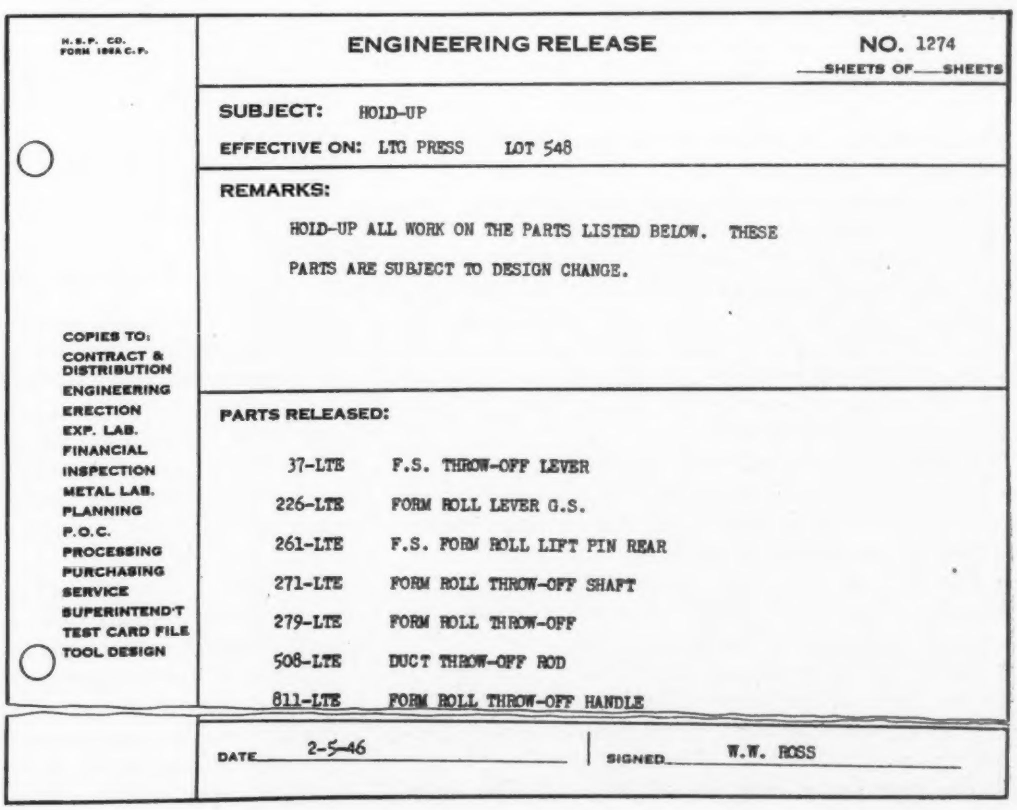
Primary function of the "Used on" column is to serve as a complete list of all the applications where any particular item is used, so that in case of an Engineering change, all B/M's or other parts listing can be corrected, or in case a change is being considered, the extent of the item's usage will act as a guide as to cost, etc.

Entries in the "Used on" column should include the numbers of all assemblies, units or parts listings where the item appears as a component. The date the posting is made should be noted in the left-hand column and the total quantity used in each application should be posted in the right-hand column.

It will be recalled that the breakdown of a machine under this record system is in the groups: Model, Units, Assemblies and Parts, in the order listed. The entry posted in the "Used on" column of a Drawing Record Card for a part is the number of the next higher group of which the part is a component. Thus, when a part appears in an assembly drawing, that assembly drawing number is the entry posted on the record card of the part and no other entry for that particular application of the part is made. Use of an assembly number or unit number on the record card of the subordinate assembly, recording the use of a unit in another unit, or in a model, is handled in the same way.

The Microfilm Record is maintained on the reverse side of the Drawing Record Card (*Fig. 6b*). New drawings and every subsequent revision are filmed. Each time the drawing is filmed the Revision number, the Roll number, the Place number on the roll, and the date of filming must be entered. "Additional Information" may be used to indicate some special information or purpose for the part, a customer's machine, references, etc.

CHANGE NOTICES: Change Notices are used to inform those who use engineering drawings and records of changes and revisions. As the same change notice is used


H.S.P. CO.
FORM 1064 C.P.

ENGINEERING RELEASE — NO. 1274
___SHEETS OF___SHEETS

SUBJECT: HOLD-UP
EFFECTIVE ON: LTG PRESS LOT 548

REMARKS:
HOLD-UP ALL WORK ON THE PARTS LISTED BELOW. THESE PARTS ARE SUBJECT TO DESIGN CHANGE.

COPIES TO: CONTRACT & DISTRIBUTION, ENGINEERING, ERECTION, EXP. LAB., FINANCIAL, INSPECTION, METAL LAB., PLANNING, P.O.C., PROCESSING, PURCHASING, SERVICE, SUPERINTEND'T, TEST CARD FILE, TOOL DESIGN

PARTS RELEASED:

37-LTE	F.S. THROW-OFF LEVER
226-LTE	FORM ROLL LEVER G.S.
261-LTE	F.S. FORM ROLL LIFT PIN REAR
271-LTE	FORM ROLL THROW-OFF SHAFT
279-LTE	FORM ROLL THROW-OFF
508-LTE	DUCT THROW-OFF ROD
811-LTE	FORM ROLL THROW-OFF HANDLE

DATE 2-5-46 SIGNED W.W. ROSS

Fig. 8 — Right — Engineering Release, official voice of engineering department, is used in place of memos or verbal orders to give data and instructions to other departments

for drawing changes as well as record changes, maximum utility was again considered. A simple form was developed, *Fig.* 7, that completely covers all the information necessary when a change is made.

This form is filled in as follows: One line is used for each part or item changed. Under "design or specification change" is listed the kind of change. Under "New Part No." is listed new part or B/M number. Under "Change" is listed the kind of change by using the V or O or # from the legend block in the lower left-hand corner.

Quantity is the same as on B/M. Under "Effective" the suitable legend is selected from the block and filled in. Part name or additional remarks are filled in under its heading. B/M reference is made as to page and line for quick reference by the change notice user. Under "Disposition", proper legend is selected for each case and filled in under each heading. If special conditions prevail the (*) is filled in and the special information outlined in "Note" block.

"Models affected" also is filled in to help users see that complete coverage is made by change notice. A short summary is written in the "Remarks" block and the file number goes in the lower right-hand corner.

If change applies to a special machine or if it is desired to keep record of the change for service, the number of the first affected machine is listed in the "Machine No." block.

Engineering Releases: These are the official voice of the engineering department. They are used to disseminate information and instructions of all kinds to other departments and are always used in place of memos or verbal instructions. They are numbered in numerical order and the original is filed in the engineering department to form a permanent record.

A special number series may be assigned to related groups of releases for cataloging or indexing. Individual copies are made for each department or person affected, and only one subject is covered on a release. A typical engineering release is shown in *Fig.* 8.

How Bill of Material Works

The following is an example of the general utility of the Harris-Seybold Bill of Material:

When management authorizes the manufacture of a lot of machines, engineering selects the proper Index B/M from the file and from this index the Unit B/M's are selected. The complete-machine B/M along with one index for each of the machines in the lot, is then white printed.

The Index and Complete B/M's then are issued on an Engineering Release to the planning department, where the manufacturing schedule and production machine load is worked out from the B/M.

The Machine Index goes into envelopes that hang on the machine to serve as a guide during erection. This index, together with a copy of all Engineering Releases and Change Notices pertaining to the individual machine, stay with the machine during its erection. After erection and testing, the index goes to the service department to form a permanent record for customer service.

After the planning department has scheduled the machine lot, the complete Machine B/M goes to the production order control department which orders the parts from vendors or writes shop orders for manufacture.

As the parts are ordered, the information is filled in on the B/M in the spaces provided. Inventory cards are posted direct from the B/M. As the parts are received into the finished stores department, assembly shop orders are written for the various assemblies. As the shop orders are issued, the B/M sheet covering the assembly is used for the finished stores requisitions, the indentions showing what parts make the assembly, thus eliminating the need for any additional clerical work in writing a requisition, as well as avoiding errors in other copying.

When all assemblies are completed and returned to finished stores, the finished-stores clerk issues all the items in the A indention only to the erection floor, thus eliminating the need to remember which parts are used in assemblies only and not on the machine. This elimination of the need for "know-how" in finished stores is extremely important where the complete manufacturing cycle is long term.

As the assemblies and parts are issued to the erection department, the B/M goes along and serves as an additional guide for erection. After the machine is erected, tested, and shipped, the B/M goes to the cost department which uses the space provided to determine the exact cost of the lot. By providing this space on the B/M, no time is lost or errors made in copying or writing additional orders or requisitions.

When a unit or assembly is required for service the service order is issued for the unit or assembly number only. The B/M then is used as the requisition from stores, for posting inventory cards, for establishing a cost of the order, for inspectors check and for packing slip. This, again, secures maximum usage and eliminates errors and time loss in copying or writing out other types of records.

A complete up-to-date file of extra printed copies is maintained by the engineering department in factory offices for quick and ready reference. Further, a complete set of Index B/M's, together with a complete set of Unit Bills, is furnished all factory branches, thus providing a quick and ready reference for information about what is standard equipment as well as providing accurate service part information.

Establishment of this system has resulted in a smoother functioning organization and actual operation has proved its ability to meet all the engineering-drawing record requirements of modern production.

Package Substations

Package substations, incorporating as a completely coordinated, ready-to-assemble unit all of the hundreds of components and parts previously obtainable on a piece-by-piece basis for installation by the customer, have been developed by the General Electric company. They are produced in any one of hundreds of arrangements in ratings up to 45,000 kva, and incorporate standard transforming, regulating, relaying, metering, and protective equipment.

Whereas, in the past, factory-assembled unit substations usually have included only the transforming, regulating and switching equipment beyond the high-voltage entrance to the transformer, the new package substations are complete in every respect. Operating data and experience gained from the first tailor-made units, developed and produced during the war, were incorporated into the present package substations.

Planetary Transmissions

. . . a design analysis

By L. A. Graham
Graham Transmissions Inc.
Milwaukee

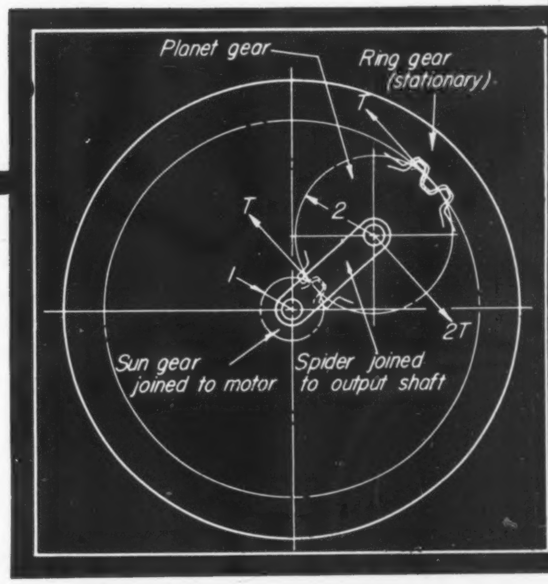
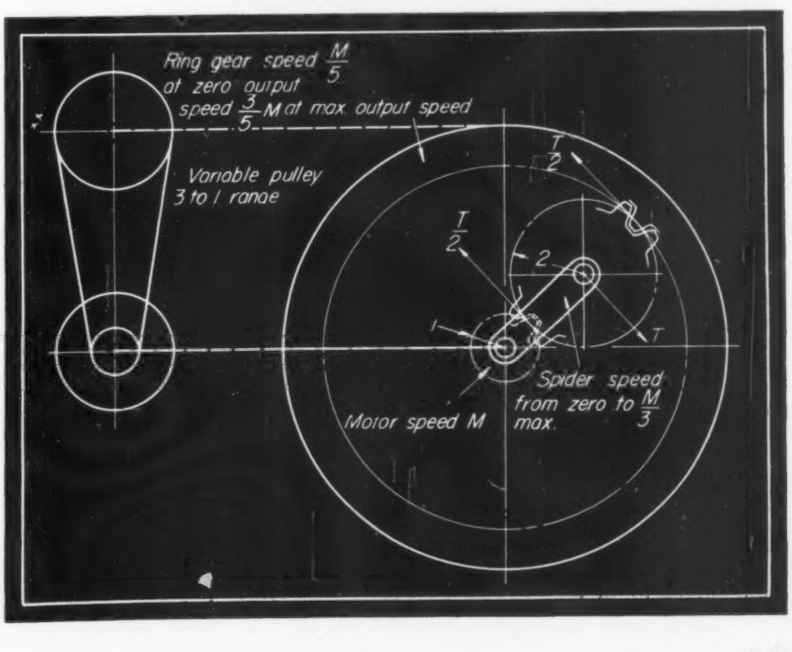


Fig. 1—Above—Simple planetary reducer with sun gear attached to motor shaft, and spider to output shaft. Ring gear is stationary

Fig. 2—Below—Differential combination of the simple planetary reducer with a variable-speed pulley

IN MANY applications of mechanical variable-speed drives, all speeds from maximum to zero are desired, including in some cases full reversal without stopping the motor. Obviously this calls for a differential of some kind. It is not easy, however, to calculate the loads and stresses on the various components of a differential. As a result, there have been some rather costly misunderstandings of what can and cannot be done with such devices. Since the Graham variable-speed transmission incorporates a differential as a basic element of the mechanism, we have had an opportunity to learn firsthand the possibilities and limitations, some of which will be set forth in this article.

Variable-speed differential drives usually involve planetary gear trains of some type and incorporate elements shown in the simple planetary reducer, in *Fig.* 1. For instance, it is well known that if the familiar variable-speed pulley of limited speed ratio—say 3 to 1 range or less—is connected to a simple planetary gear set as illustrated in *Fig.* 2 so as to utilize differential speeds properly, it is possible to obtain a speed variation in the output shaft of the gear train, from maximum to zero and reverse.



Several variable-speed drives incorporate this principle and are in successful use. Sometimes, however, exaggerated claims have been made for this arrangement. For example, the Patent Office abounds with inventions for automobile transmissions of this general character in which the variable part of the drive is made of relatively small dimensions even though automotive transmissions must have essentially a constant power characteristic. This is done on the theory that only a small part of the power need be "by passed" through the variable pulley, the bulk of the power being presumably carried through the planetary gearing. Actually, in

order to make possible speeds down to zero, the variable part of such a differential drive, even for constant torque, must be large enough, if the particular arrangement shown in *Fig.* 2 is used, to carry *more* than the external load horsepower, rather than a small fraction of it, as will be shown later.

To make this clear, let us consider first a simple *non-variable* planetary gear system, *Fig.* 1, which delivers only a single speed and is commonly used for some makes of gear-motors in ratios up to about 10 to 1. A sun gear joined to the motor shaft, a planet gear (usually there are 3 of these), a stationary ring gear, and a spider keyed to the output shaft make up the reducer. If the sun gear radius

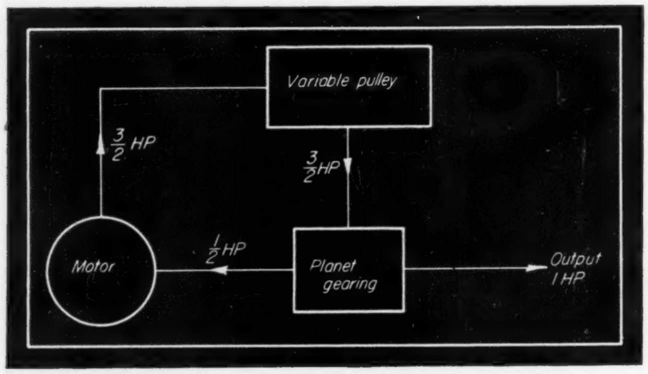
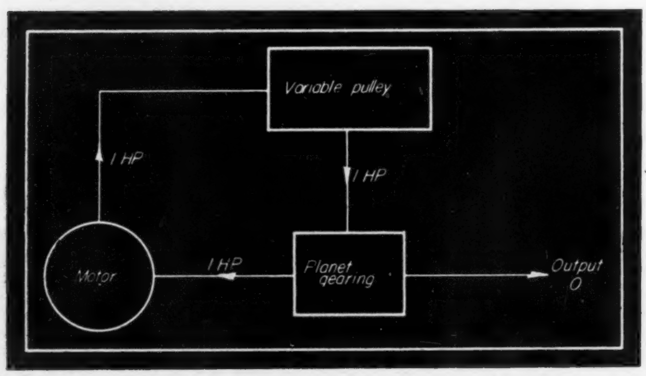


Fig. 3—Above—Power flow in variable-speed transmission at maximum output speed for one horsepower output. The unit must be designed for 50 per cent more power than is required at the output shaft

Fig. 4—Below—Power flow at near zero speed with output torque twice its value at high speed. Circulating power is 1 horsepower for double torque near zero speed



is taken for convenience as unity and the planet gear radius as 2, it is evident that for a motor torque equal to T, the tooth pressure between planet gear and both sun and ring gear is also equal to T. The planet bearing pressure (neglecting losses) equals $2T$ and the output torque is $2T$ times 3 or $6T$. The reduction is thus 6 to 1, and in the general case it may similarly be shown that the reduction equals the ratio between ring gear and sun diameter (or ratio of number of teeth in each) plus 1.

Although this makes a compact, concentric single-speed reducer of wide utility for moderate reductions, a frequently-overlooked disadvantage of such a gear unit is that the ring gear, although it does not turn, is subject to a combination of tooth pressure and relative gear speed which result in the same amount of frictional loss as if the horsepower represented by the product of the speed and the force at that point were *productive*, rather than "internal" or nonproductive. For 1 hp output this product would equal 5/6-hp in the example chosen. This loss, incidentally, is exactly comparable to the losses at the stationary rail below a locomotive driving wheel. Further losses incidental to this type of reducer arise, of course, at the two bearings required for each planet and, since these particular losses are not present in a simple gear train of the same 6:1 reduction, the planetary reducer usually has a correspondingly lower efficiency. Nevertheless, a properly designed unit of this kind may have an efficiency well over 90 per cent, and does a thoroughly satisfactory job, especially in the lower horsepowers.

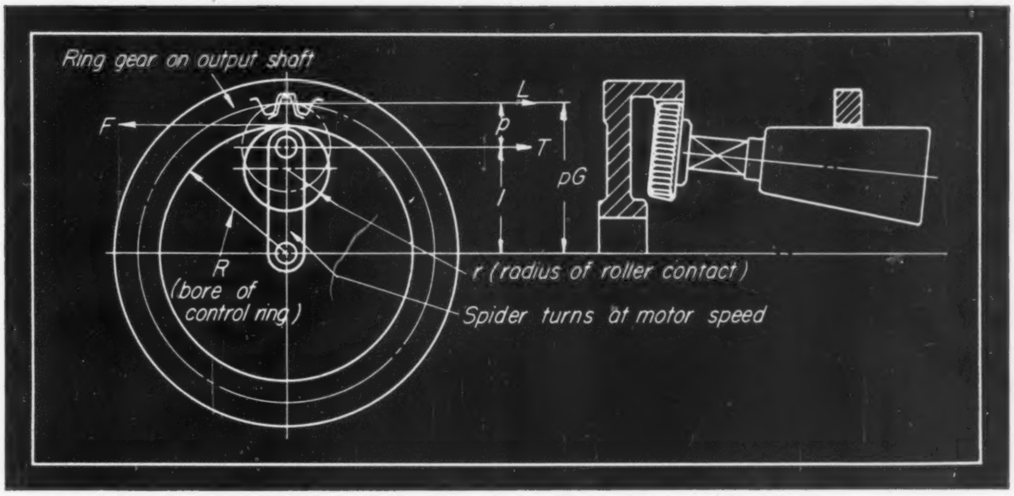
Variable Speed Obtained By Rotating Ring Gear

This same planetary gear unit may be converted into an infinitely-variable speed transmission of wide range by joining it to a variable pulley drive, in the manner shown in *Fig.* 2. The sun gear is joined direct to the motor shaft as before and the spider to the output shaft but the ring gear, instead of being held stationary, rotates at a variable speed over a range of 3 to 1, by being joined to the driven shaft of a variable pulley of that ratio. In some forms, there is a speed reduction between motor and sun gear, and between pulley and ring gear, but the characteristics here discussed remain the same. Since the prime object of such an arrangement is obviously to extend the range from 3 to 1 to zero output speed, the conditions necessary to obtain zero speed of the load shaft will be considered first.

With the spider speed zero, the planet is evidently an idler, and the drive corresponds to the internal gear drive used on some truck axles. This means that a speed must be imposed on the ring gear equal to 1/5 of motor speed and in a direction opposite to the motor rotation. Since the variable pulley in this case has a range of 3:1, its speed for maximum output speed of the combination (assuming there is no reverse range in the transmission) must either be 3/5 of motor speed or 1/15 of motor speed. The higher of the two gives a correspondingly higher top speed to the transmission, and since this is usually an advantage in variable drives (high top speed means lower torque requirement and smaller parts), that figure will be used in this example.

Referring to *Fig.* 2, the spider speed S, corresponding to motor speed M and ring-gear speed $3/5M$, is $M/3$, which means that such a transmission will deliver speeds from a maximum of 1/3 of motor speed down to zero (note that the 2π factor would cancel out in the computations). The planet bearing pressure is now T instead of $2T$ as in *Fig.* 1, since the output speed is doubled, so that the pressure between planet gear and both sun and ring gears is half what it was in the single-speed arrangement. However, since the ring gear now turns, the power required at the tooth contact, which is equal to $3M/5 \times 5 \times T/2 = 1.5$ hp, must actually be transmitted through it and furnished to it from the variable pulley. The flow of power is illustrated schematically in the diagram, *Fig.* 3. The excess power which comes back to the motor is sometimes referred to as a "circulating power". This circulat-

Fig. 5—General conditions for Graham variable-speed drive



ing power is the price paid in this particular arrangement for wide speed range and it is to be noted that, in the example chosen, it definitely calls for a variable pulley large enough in dimension to take 50 per cent more power than the maximum useful output of the entire transmission. This effectively quashes all notions that the variable part of such a combination can be made small because it "by passes" most of the power. Incidentally, the only way such a device can be used to bypass a substantial part of the power is by designing it for a *small* speed range only and then the differential is unnecessary.

Power Characteristics Near Zero Speed

Needless to say, a drive with speed range down to zero cannot have a constant power characteristic. If the torque output at near zero speed is required to be no greater than at top speed, the power delivered to the ring gear will be $M/5 \times 5 \times T/2 = 0.5$-hp. The power flow will be as in *Fig.* 4, in which it is evident that this ½-hp is entirely a circulating power, since practically no useful power is delivered when the output speed is near zero. If it is desired to obtain double the torque at near zero speed, as compared to top speed, 1 hp must circulate even though no power is delivered; if the torque is to be tripled, the circulating power increases to 1½ hp. Again, however, it must be stated that with modern design this power does not make the losses prohibitive but merely requires that the parts in both sections of the combination—the variable pulley and the planet gearing—be large enough to handle the speed and loads.

It is true, however, that the familiar arrangement so far discussed is needlessly complex and inefficient, chiefly because of the two separate units which it involves and the power required to circulate between them. There is a way to avoid these drawbacks, while retaining the advantages of the differential, and secure a simple, compact, efficient, low-priced infinitely-variable speed transmission with speeds from top to zero, plus reverse if wanted. The general arrangement, *Fig.* 5, for accomplishing this was originated over 50 years ago and is applied in the Graham drive. Instead of belts it makes use of metallic rolling contact.

There are four other vital differences from the planetary gear previously analyzed. First, the planetary is of the so-called compound type, similar in principle to that used in the early Ford selective transmission. Second, the sun pinion is eliminated. Third, the variable part of the drive is inbuilt as part of the gearing and is not external to the gearing as in the previous example. Fourth, by these means all circulating power is avoided, and the gear corresponds closely in efficiency and general characteristics

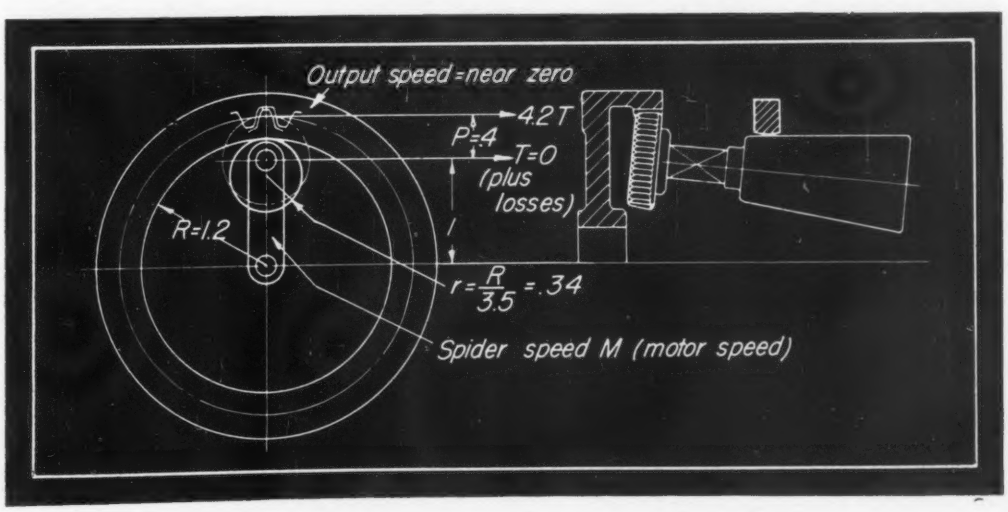


Fig. 6—Typical conditions in transmission for doubled torque at near zero speed

to the single-speed planetary discussed at the beginning of this article.

In this drive, *Fig.* 5, the spider instead of being joined to the output shaft is direct-driven by the motor and, being fixed to the motor shaft, it turns at all times at motor speed. Each planet member carried by the spider (there are onlv two planets in this system) comprises two gears on a common shaft, one a tapered roller contacting under pressure a stationary control ring and the other a toothed pinion which meshes with and drives a ring gear joined to the output shaft. The roller derives its rotative speed from its tractional contact with the stationary ring. This rotative speed is in the opposite direction from the motor rotation and therefore subtracts from motor speed to give the output speed of the transmission. The planet member is on an inclined axis the inclination of which is equal to the taper of the roller, permitting the cylindrical ring to be moved lengthwise along the outer edge of the two rollers to change the speed.

Speed and torque characteristics of this drive are readily analyzed by the same method used on the planetary transmission. Assuming the motor torque radius (distance from center line to center of planet pinion) to be unity as in the previous example, the motor torque is T as before. If the ring gear tooth pressure is L and its pitch radius is pG, then LpG is the load torque and the ratio between LpG and T determines the ratio between speeds of output and input shafts. It is evident, too, that the difference between these torques must equal the tractional reaction torque FR, so that $T = LpG - FR$ from which is derived the speed ratio $T/LpG = 1 - FR/LpG$. However, since the moments around the planet axis must equal 0, $Lp = Fr$ or $F = Lp/r$ from which the speed ratio is

$$1 - \frac{LpR/r}{LpG} = 1 - \frac{R/r}{G}$$

which may be conveniently written as speed ratio = 1 minus the ring : roller ratio at the tractional contact divided by the gear ratio, expressed in terms of number of teeth in each.

From this simple relationship it is evident that the output speed becomes zero when the two ratios are equal, so that to obtain zero speed in this transmission the control ring, *Fig.* 6, is moved along the outside of the two tapered planet rollers until this equality is reached.

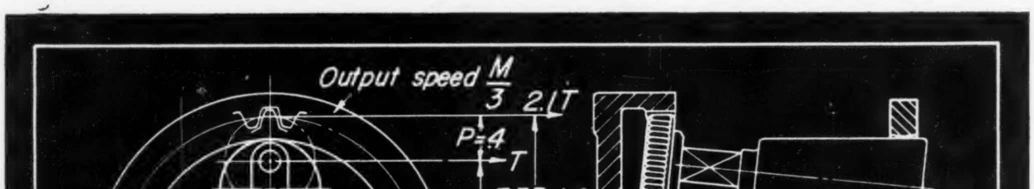


Fig. 7 — Typical conditions for maximum output speed of unit

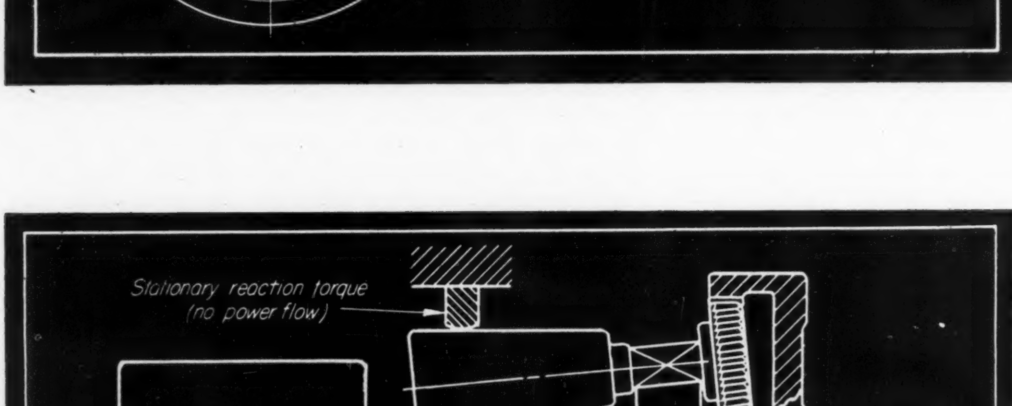


Fig. 8 — Schematic of power flow in Graham transmission

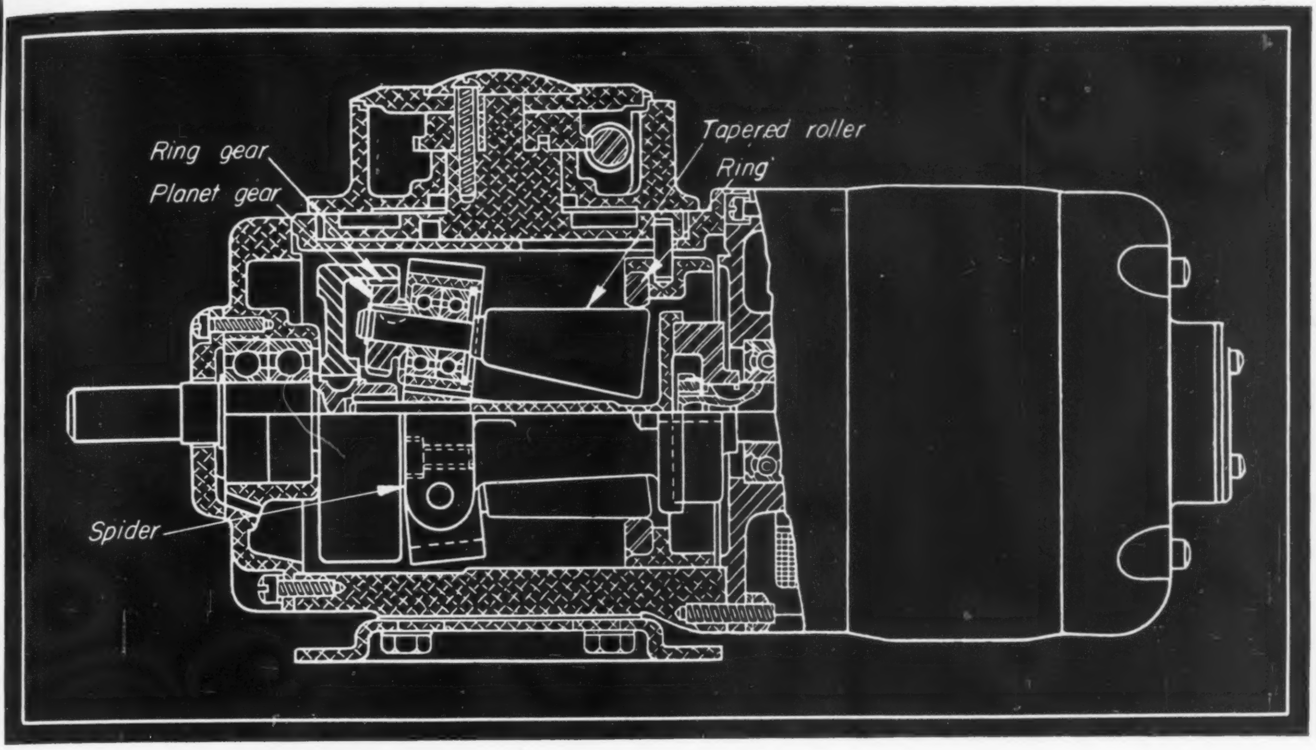


Fig. 9—Sectional view of transmission unit showing hinged mounting for rollers and minimum number of bearings

When full power is transmitted at top output speed, R/r in the Graham drive is designed to be approximately 2.3, *Fig.* 7, and since G is usually about 3.5, maximum output speed equals $(1-2.3/3.5)$ or about 1/3 input speed, as in the previous example. Taking R as 1.2, r becomes 0.52 at top speed and 0.34 at zero speed, which shows that an increase in roller dimension of only 50 per cent is necessary in such an arrangement to give every output speed from 1/3 of input to zero. Where full speed in reverse is wanted, it may be shown that for the same ring and roller dimensions G is reduced to 2.9 (pinion and gear are made larger), and all speeds are obtained from about 1/5 of motor speed forward, through zero, to an equal speed in reverse.

Has No Circulating Power

As shown in *Fig.* 8, there is no "circulating power" in this arrangement since the control ring is stationary. Therefore the losses are similar to those present in nonvariable planetary gearing. Actually at top output speed and full load the measured efficiency of a ¾-hp unit is in excess of 85 per cent, which is about 5 per cent less than the efficiency of good planetary speed reducers. This would indicate that the differential-planetary variable drive is by no means beset with limitations not found in other mechanisms. Obviously, where speeds go down to zero, such a drive cannot have a "constant-power" characteristic over the full range, but the design may be such that the torque not only is maintained over the entire range to zero but also is doubled at low speeds, which answers most common requirements for variable-speed transmissions.

Most planetary drives suffer from a surfeit of costly ball bearings. Even the simple single-speed planetary, *Fig.* 2, requires ten ball bearings—two for the input shaft, two for the output shaft and two each for the three planetary gears—ten in all. If a separate variable pulley is used, this adds at least four more bearings. On the other hand, in the Graham transmission with built-in motor, *Fig.* 9, there are no additional bearings for the spider which is carried by the motor bearings; there are the two conventional bearings for the output shaft and beyond this only a duplex bearing at the small end of each of the two rollers. There are no bearings at the large end of the roller and this economy was made possible in the current design by hinge-mounting the rollers. The radial load is thus carried by the duplex bearings plus the control ring, and the tangential load is relatively so slight as to be fully supported (cantilever fashion) in the duplex bearings. By avoiding the need for bearings at the large end of the rollers, one-third more has been added to the useful length of the roller, with corresponding increase in power capacity for a given size of housing.

Another recent design economy consists in controlling the speed by engaging the control ring at one point only which is done through a grooved cam having the shape of a logarithmic spiral. This gives the added advantage of an extremely useful linear relation between control movement and speed. A still further saving comes from the use of many die castings instead of gray iron sand castings, greatly reducing the amount of machining.

PLASTICS known as Infropake transmits visible light only, absorbing infrared and ultraviolet. It transmits approximately 85 per cent visible light with practically no ultraviolet transmission between 200 and 400 millimicrons and about 15 per cent infrared between 750 and 1500 millimicrons. Experimental evidence indicates very little transmission of frequencies ranging between 1500 and 2000 millimicrons.

Full-Flow Filt

. . . provides maximum protection against wear by delivering only clean oil to the working parts of an engine.

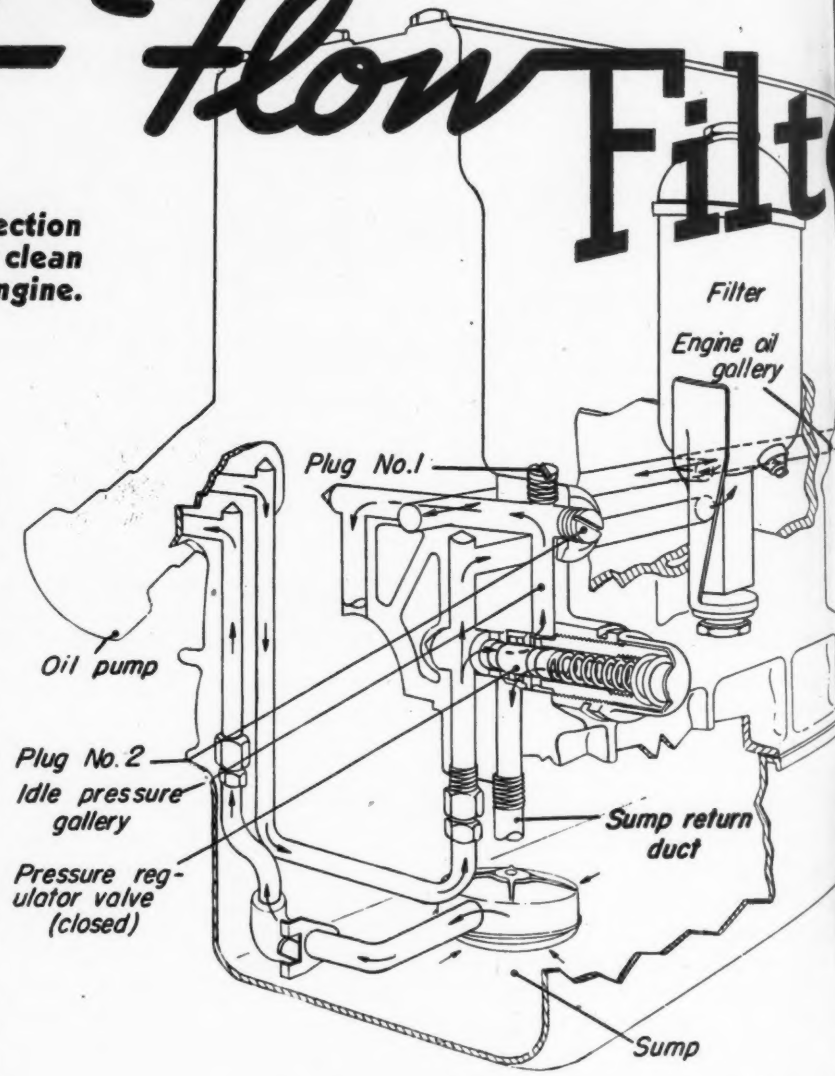


Fig. 1—Pad-mounted full-flow filter on an 8-cylinder engine. Oil is circulating through idle pressure gallery

TO KEEP an engine well lubricated is an unqualified advantage to be sought which takes more than merely a plentiful supply of oil. In order to insure a long and efficient power-plant life it is equally important that dirt, abrasives and other foreign matter be kept away from the vital working parts of the engine, particularly the bearing surfaces and cylinder walls. Because lubricating oil, forced to these spots, can be and often is the means of bringing in wear-producing materials, there must be some way to assure that only clean oil circulates through the system.

Generally, oil filters of the conventional by-pass type handle only a portion of the total amount of oil being delivered from the oil pump. Now, however, with a new system used on the current Chrysler 6 and 8-cylinder cars, another forward step has been taken in the protection of engine life by the use of a full-flow oil filter, *Fig.* 1. With this filter, the normal operating condition is for all oil from the pump to pass through the filter on its way to the engine lubricating system.

There is no question but what a by-pass filter does an excellent job of filtering the oil that goes through it. In fact, this type of filter has been brought to a point of high efficiency from a filtration standpoint. The difficulty is that a conventional filtering system, designed to operate on a by-pass circuit, loses something of its effectiveness owing to the fact that the clean oil is delivered not direct to the engine, but back to the reservoir in the crankcase. Thus, oil pumped to the engine is a mixture of the filtered and unfiltered oil. It is probable that, over a period of time, all of the oil passes through the filter but, on the other hand, oil going direct to the engine is never completely clean.

With the use of a full-flow filter, it is possible for all of the oil going to the engine to pass through the filter thus delivering only clean oil into the engine oil galleries. In order to assure adequate lubrication throughout the engine at starting or when idling with the oil at a high temperature, a special pressure-control valve automatically permits variations in the circuit when required. Under normal operation, however, the engine enjoys the important advantage of having all of the oil go through the filter before being circulated in the engine.

In operation, the full-flow filter is simple, since most of the time the oil is pumped through the filter and into the oil galleries of the engine block. However, there are two variations which must be provided—one is when insufficient oil is going through the filter, and the other when more oil is supplied than is required by the engine or than the filter can handle. Accordingly, a pressure control valve is included in the oil circuit. In order to provide for below normal, normal, and above normal oil pressures, it has three positions—closed, intermediate and full open.

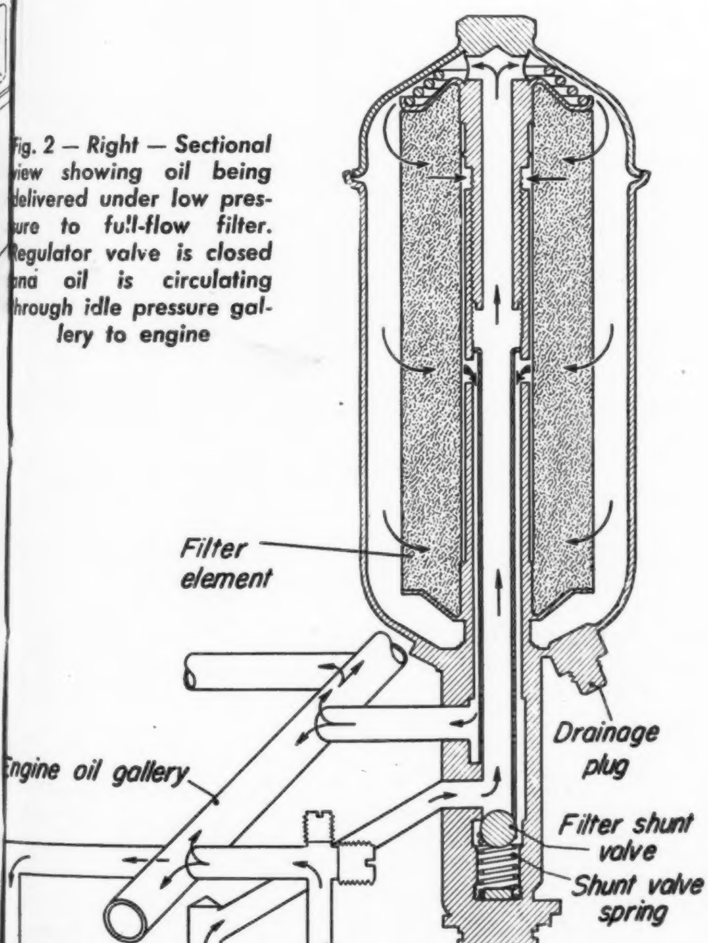
In the full-flow system, *Fig.* 2, the pressure valve is shown in closed position, as it would be when starting the engine. If the car had been standing for some time, some of the oil would have drained down out of the filter, which means the engine would operate for an appreciable number of seconds before the filter was filled with oil to the point where it would deliver the normal quantity of oil under pressure into the engine block. However, the

ting

engine has a plentiful supply of oil during this brief starting period, because with the control valve held in the closed position by its spring, some of the oil from the pump can go direct to the engine, through the idle pressure gallery, as shown by the arrows. The major supply of oil still goes up into the filter, filling it and starting the delivery of oil from the filter into the engine oil gallery and thence to the various engine parts.

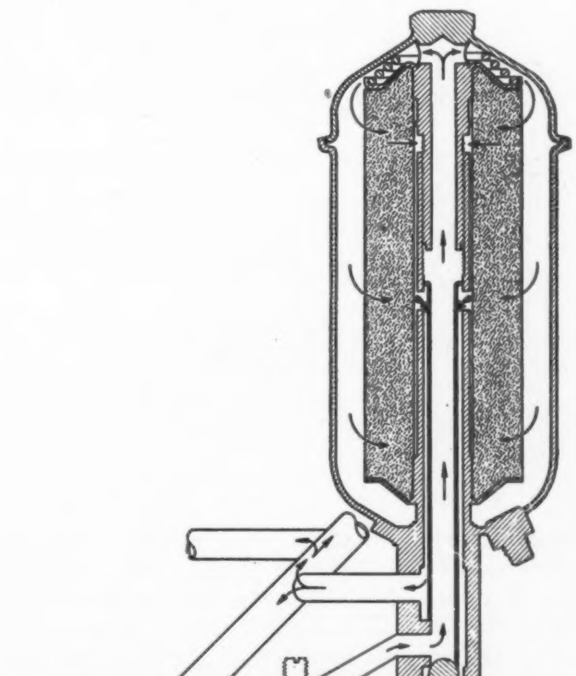
In a few seconds oil pressure builds up to the point where the force of the oil on one end of the control valve, is greater than the force exerted by the spring on the other end. This moves the valve into the intermediate position as shown in *Fig.* 3. Since it is no longer needed, the idle pressure gallery to the engine is now completely closed, and all the oil from the pump now goes to the filter, from which it is delivered under pressure into the engine block. The time at which the filter goes into full action is indicated by the oil pressure gage on the instrument panel. Since the pressure is taken at a point beyond the filter, the needle does not respond when the engine first starts, but only as the filter takes over the job of being the sole supplier of oil for the engine. As has already been noted, the engine receives an ample supply of oil from the start, so this delay in the registering of oil pressure on the instrument panel does not indicate danger to the engine.

At high speeds the oil supply from the pump ordinarily exceeds the needs of the engine and capacity of the filter so, as the oil pressure increases, the control valve continues to move until it reaches the third, or wide-open position, *Fig.* 4. This opens a line to the crankcase, which allows the surplus oil to return without going to the engine or building up excessive pressure



ig. 2 — Right — Sectional iew showing oil being elivered under low pressure to full-flow filter. egulator valve is closed nd oil is circulating hrough idle pressure gallery to engine

Fig. 3—Below—View of filtering system with regulating valve at intermediate position. Idle pressure gallery is blocked and all oil is delivered through the filter to the vital engine parts



in the filter element or associated equipment.

Obviously, if the filter element should become clogged, and the owner fail to replace it with a new one, the point would eventually be reached where not enough filtered oil would be delivered to the engine. Since it is deemed better to circulate unfiltered oil for a temporary period, rather than burn out bearings or score cylinders, a spring-loaded shunt valve is provided in the base of the filter to take care of this condition.

Again the action is simple and automatic, being controlled by oil pressure. As the filter becomes clogged, the pressure of the oil leaving the filter begins to drop. When a difference in pressure of about 17 pounds between incoming and outgoing oil is reached, the spring-loaded shunt valve opens, allowing oil to flow from the filter intake line into the engine block bypassing the filter. Advantage in connecting the oil-pressure gage beyond the filter immediately becomes apparent, since it shows the clogged condition of the filter by registering a definite drop in indicated pressure. Accordingly, a drop in operating pressure from around 50 pounds to around 35 pounds, indicates that the filter is no longer functioning, and that the element should be replaced.

From a servicing viewpoint, two features of the pressure-control valve are noteworthy. First, as indicated by arrows on the drawings, provision is made for oil to drain from behind the valve, so that its operation will not be stopped bv trapped oil. Second the valve is entirely symmetrical, so it will work equally well installed either end first, or turned in any direction.

While filters designed for 6-cylinder cars differ somewhat from those on the 8, their efficiency and operation are the same. The use of a pad mounting on the 8-cylinder block permits internal arrangement of the filter oil passages, while external lines are required on the 6-cylinder installation. Filters used contain an element consisting of cotton yarn spirally wound on a wire mesh core, *Fig. 5*. The direction of the spiral is reversed for each layer, thereby making it practically impossible for oil to "channel" through the element. Paper filter elements, of equal efficiency, are optional equipment.

It is evident that the full-flow filter affords several important advantages. It insures that except for the very first few seconds of starting, or when idling with high oil temperature, all the clean and filtered oil is delivered direct to the engine rather than first being mixed with the sump supply of contaminated oil. Should the filter become clogged, an immediate warning is given and at the same tme the oil supply to the engine is maintained. Finally, the controls of the system are simple and automatic.

While no firm comparative figures can be given to show the extra protection given an engine equipped with a full-flow filter system, accelerated laboratory tests definitely prove the superiority of the full-flow filter over the by-pass type, in reducing engine wear and thus adding to useful engine life.

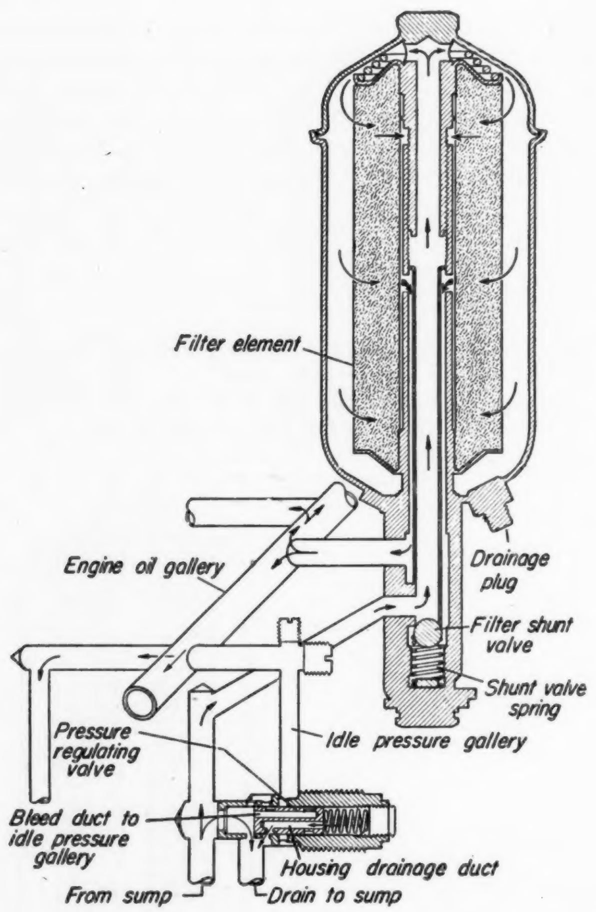


Fig. 4 — Left — Filtering system with pump delivering oil to filter at high pressure. Regulator valve is at wide-open position allowing excess oil to return to the sump

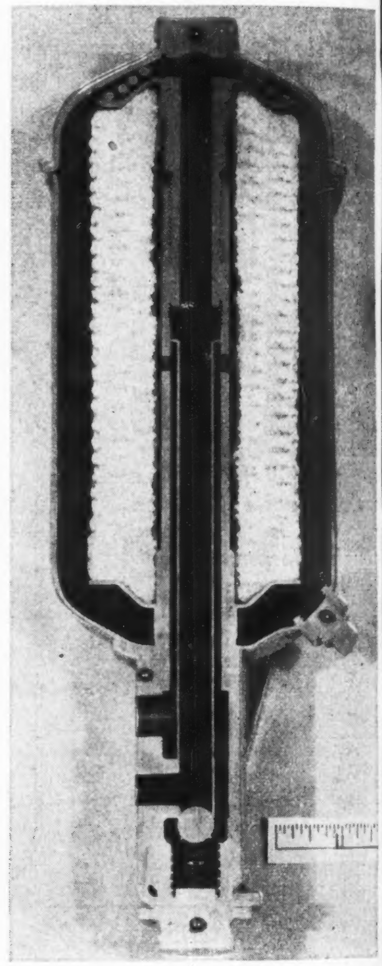
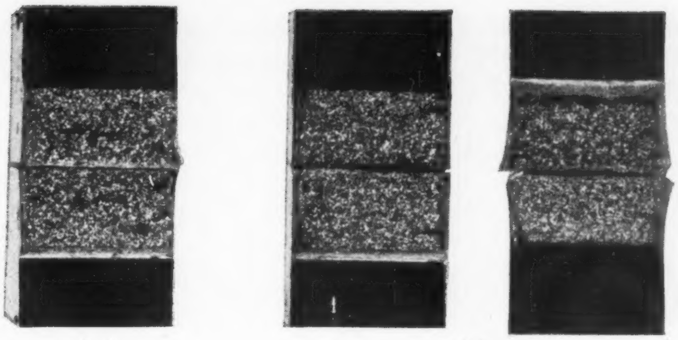


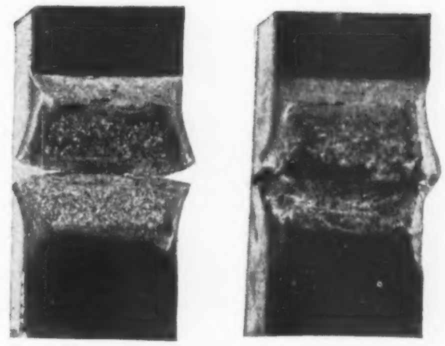
Fig. 5 — Right — Cross sectional view of typical filter design



(a) (b)

Fig. 1—Appearance of the fractures of spheroidized SAE 1030 steel tested at temperatures covering the transition (brittle failure) zone. (a) Below transition zone; (b) In transition zone; (c) Above transition zone

MILD Steel



(c)

... may be subject to unexpected shock failure. Corrective measures lie in proper relationship of cold working to annealing

By H. Markus and D. F. Armiento

Metallurgists
Laboratory Division
Frankford Arsenal, Philadelphia

IN THE DESIGN of steel parts such as machine members, it has been generally accepted that cold working of low-carbon steels, such as drawing of shells, has no adverse effect upon the toughness unless of an exceptional degree and accompanied by critical annealing. However, in recent years it has been observed that grain coarsening such as seen in *Fig.* 1 may occur in steels such as SAE 1030, under certain moderate combinations of straining and annealing, with a particularly unfavorable effect upon toughness in the relatively mild temperature range from 0 F to 50 F. To determine the conditions leading to this somewhat unpredicted weakening, a series of tests were run by this laboratory on SAE 1030 steel in an attempt to learn what fortuitous circumstances lead to the exceptional grain growth, and what corrective measures might be taken.

MATERIALS: The material used in the studies was spheroidized aluminum killed SAE 1030 steel and was obtained in the form of disks, 0.690 inch in thickness and 9.225 inches in diameter. The chemical composition of the steel is shown in TABLE I.

After suitable cold reductions were made, standard Charpy impact specimens were ground from the 1¼ inch wide strips. Impact tests were made with a Riehle impact machine having a capacity of 60 ft-lb and a striking velocity of 18.1 ft/sec. In the low-temperature tests the specimens were immersed in an ethyl alcohol and dry ice bath or Varsol and dry ice bath, while for the room-temperature and high-temperature tests a water bath was employed.

GRAIN COARSENING: It has been accepted generally that abnormal grain growth with resulting decrease in toughness occurs in steels containing less than 0.14 per cent carbon only when critically strained and annealed. However, in recent years it has been observed in the fabrication of deep-drawn articles from disks that grain coarsening may occur in SAE 1030 steel under certain

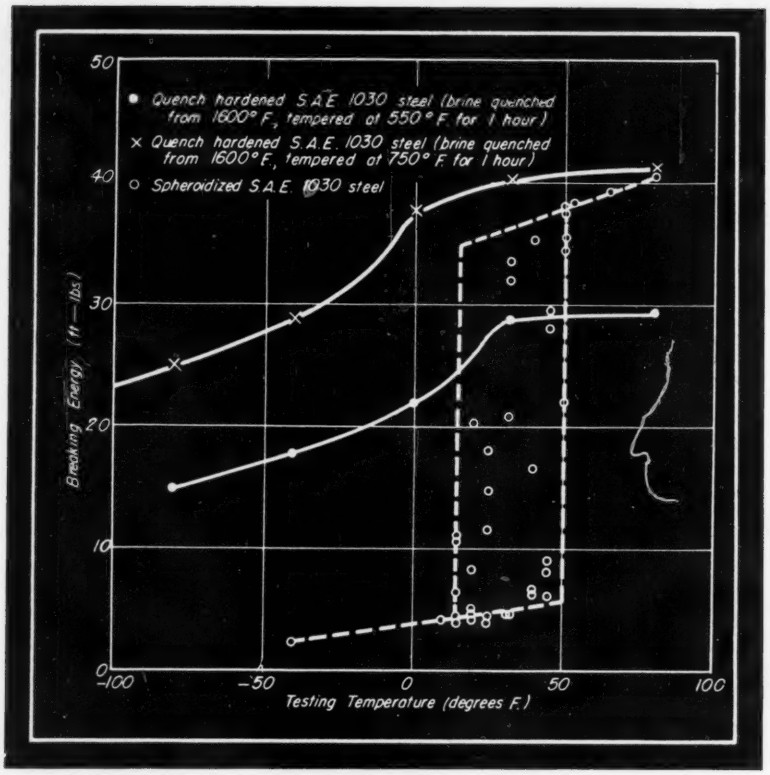


Fig. 2—Effect of testing temperature on the Charpy notched-bar impact strength of quench-hardened and spheroidized SAE 1030 steel

moderate conditions of straining and annealing.

Methods: A portion of the spheroidized material was converted into pearlitic by a double-normalizing treatment in a controlled-atmosphere furnace. This treatment consisted of heating to 1850 F, soaking 2 hours, and air cooling, then reheating to 1550 F, soaking 1 hour, and air cooling. Cold rolling of the spheroidized and pearlitic materials was conducted after cutting strips 1¼ inch by 0.690 inch from the steel disks parallel to the rolling direction. These strips were cold rolled in the original rolling direction approximately 0.010 to 0.012-inch per pass.

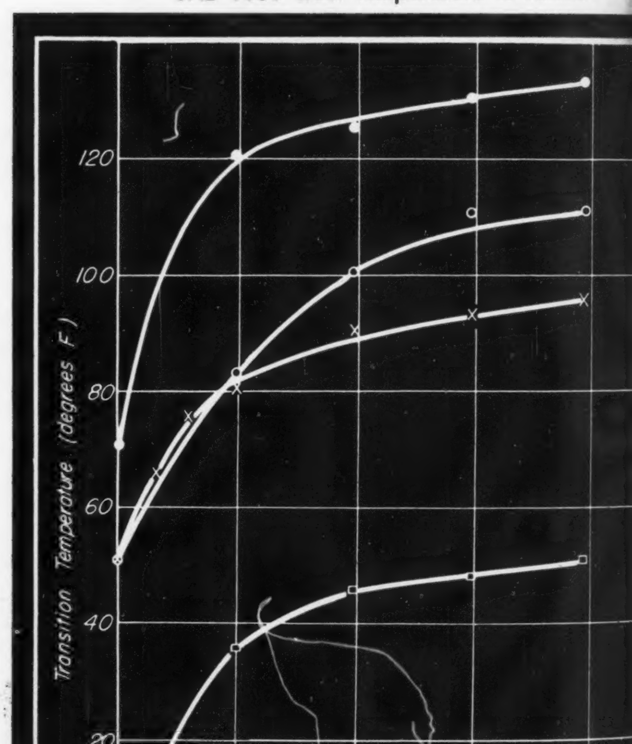
Table I
Composition of Test Steel

Element	Chemical Composition (per cent)
Carbon	0.2
Manganese	0.64
Phosphorus	0.020
Sulphur	0.026
Silicon	0.01
Copper	0.06
Aluminum	0.01

To determine the amount of cold deformation required for grain coarsening in SAE 1030 steel, strips were cold rolled 0 to 20 per cent (by 2½ per cent increments) and annealed at 1275 F for five hours. It was found that the critical degree of cold working was 10 per cent for the spheroidized material and 12½ per cent for the pearlitic material. Cold work in excess of this value resulted in reduced grain size and, in some cases, increased toughness.

Impact Properties: Spheroidized and pearlitic steel has been described by Sergeson & Poole*. The test data showed a transition temperature range wherein a tough to brittle type of failure was observed as the testing temperature was lowered. Preliminary Charpy impact test data obtained on SAE 1030 steel indicated that a similar loss in toughness was noted as the testing temperature was lowered. This is illustrated in *Fig.* 2. The highest temperature at which brittle fracture still persisted was found to be 50 F. An interesting phenomenon may be noted in the data in *Fig.* 2 in the temperature range of 15 to 50 F. In this range it is observed that both tough and brittle fractures were obtained and that the values varied from 4 to 38 ft-lb. The appearance of the fracture of the specimens which ruptured in a tough manner was primarily fibrous although some granular material was present while the fracture of samples which failed brittlely appeared granular. A combination of fibrous and granular structure was obtained with those specimens which broke with intermediate values. The ratio of fibrous to granular structure appeared to be related to the breaking energy of any particular speci-

Fig. 3—Effect of cold working on the transition temperature of fine and coarse-grained SAE 1030 steel temperature is raised



* *Proceedings ASTM* Vol. 36, Part I, Page 132, 1936.

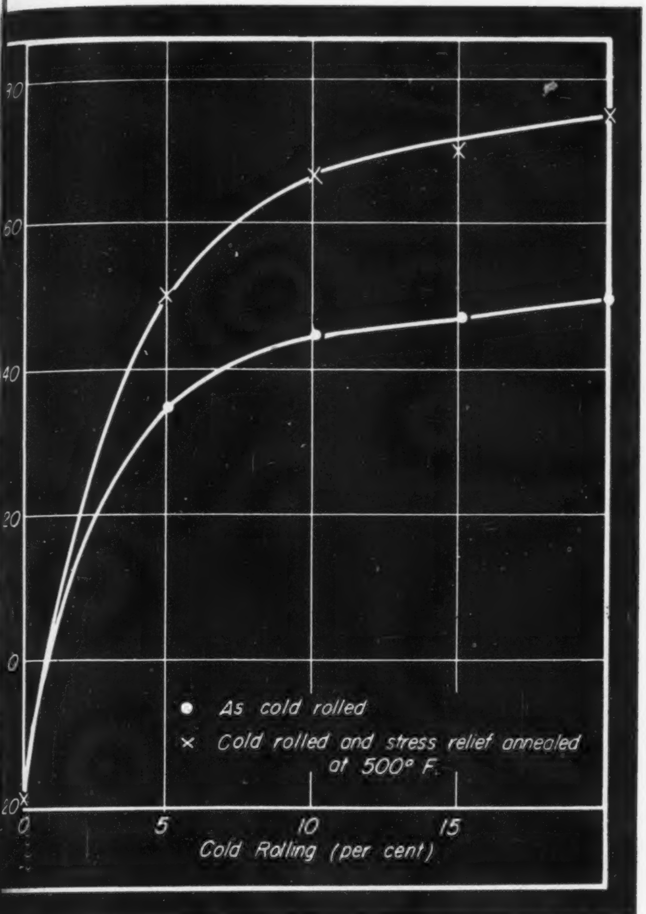


Fig. 4—Effect of stress-relief annealing at 500 F on the transition temperature of fine-grained pearlitic SAE 1030 steel. Annealing raises transition temperature

men. Thus a specimen which showed a large amount of granular but some fibrous structure required slightly more energy for failure than the completely granular specimens. These structures are illustrated in *Fig.* 1.

In order that the toughness of the steel treated in various manners could be properly evaluated, Charpy impact test specimens were prepared and tested at a series of temperatures. The highest testing temperature at which brittle failure still persisted was selected for rating the toughness of the material. This temperature has been designated as the transition temperature. For the various test investigations a fine and a coarse grain steel were selected from each of the spheroidized and pearlitic samples previously prepared for the grain coarsening experiment.

The effect of cold working on the Charpy impact toughness of the fine-grained spheroidized material is illustrated in *Fig.* 3. As little as 1½ per cent cold working raises the transition temperature 15 F while 20 per cent cold working increases the transition temperature 45 F. This effect is similar for pearlitic material. In this figure there is also illustrated the effect of microstructure, grain size, and degree of cold working on the toughness of the steels being considered.

The Charpy impact toughness as influenced by stress relief annealing of the cold worked material is illustrated in *Figs.* 4, 5 and 6. Annealing the cold-worked specimens between 400 F and 800 F elevated the transition temperature considerably and at 500 F the maximum effect was observed, indicating the negative effect of the heat treatment. Hardness values for the steels are shown in Table II.

Quenched Steel: The experience at Frankford Arsenal with SAE 1030 steel indicated that the toughness of quench-hardened steel was superior to that of the spheroidized or pearlitic cold-worked steel. Therefore, a series of

Table II

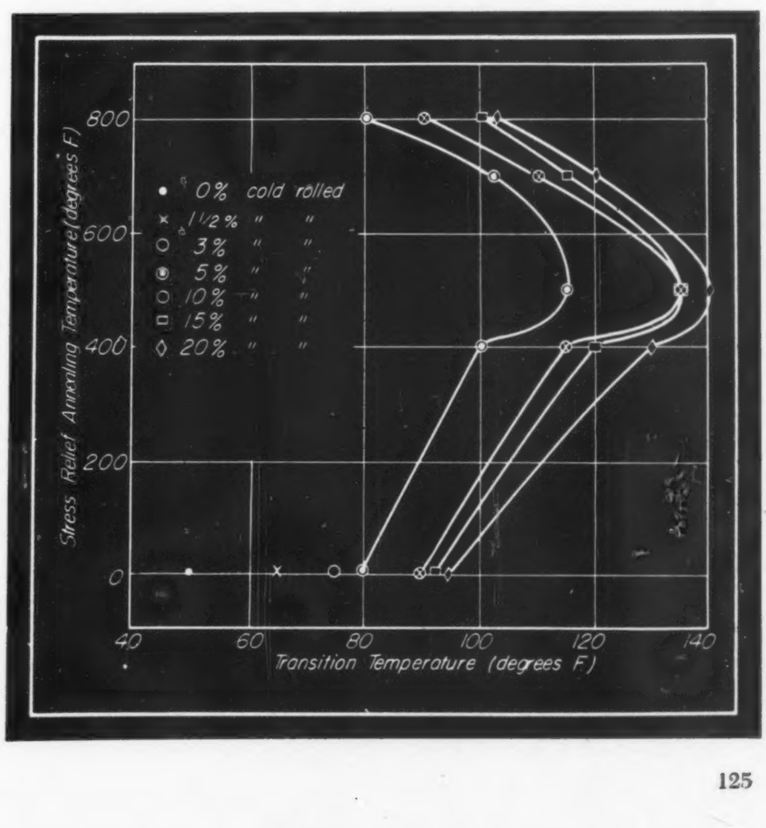
Rockwell B Hardness of Cold-Worked SAE 1030 Steel

Per Cent Cold-Worked	Fine-Grain Steel Annealed One Hour				Coarse-Grain Steel Not Annealed	
	Spheroidized		Pearlitic		Spheroidized	Pearlitic
	0 F	500 F	0 F	500 F		
0*	61.5	....	68.0	65.5	66.5	65.5
5	75.0	77.0	78.0	78.0	76.5	73.0
10	79.5	81.0	83.5	81.5	80.5	75.0
15	81.0	84.5	85.5	87.5	83.5	83.0
20	81.5	86.0	88.5	89.5	84.0	86.5

Charpy impact specimens were prepared in the quench-hardened condition for tests so that comparative data could be obtained. The heat treatment employed consisted of austenitizing at 1600 F, quenching rapidly into agitated 10 per cent brine solution and tempering at 750 F or 550 F for one hour; resultant rockwell C hardness is 38 and 40.

The Charpy impact test results are illustrated in *Fig.* 2 and show the usual behavior for this material. It may be noted that the transition zone differs considerably from that for the spheroidized specimens. There is no sudden drop in Charpy impact values but rather a more gradual lower-

Fig. 5—Effect of cold working and stress-relief annealing on the transition temperature of fine-grained spheroidized SAE 1030 steel



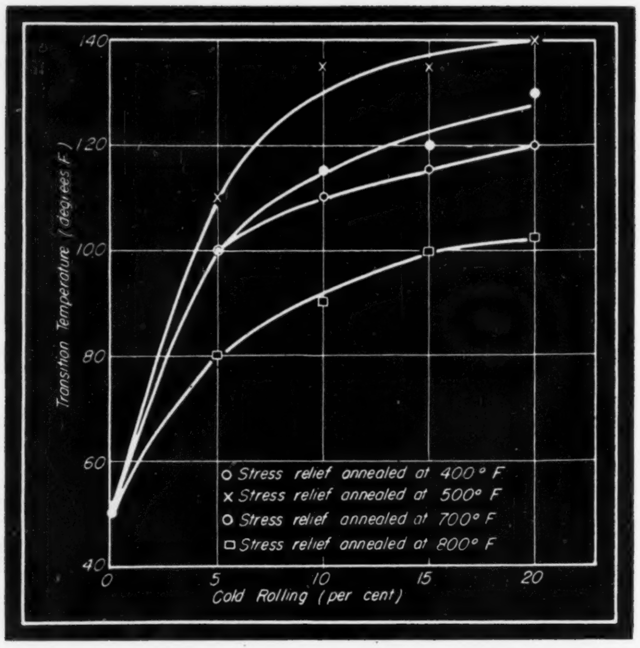


Fig. 6—Effect of cold working plus stress relief annealing on the transition temperature of fine grained spheroidized SAE 1030 steel. Working raises transition temperature

ing of the values with decrease of testing temperature. For a given testing temperature there are no instances where the Charpy impact values are highly divergent. Appreciable improvement in toughness is obtained by the quench hardening treatment. Even at the lower temperatures (-40 F) the quench-hardened specimens possessed relatively good toughness.

CONCLUSIONS: Abnormal grain growth occurs in both spheroidized and pearlitic SAE 1030 steel. Conditions used for producing abnormal grain growth are: For spheroidized SAE 1030 steel, cold roll 10 per cent then anneal at 1275 F for 5 hours, while for pearlitic SAE 1030 steel, cold roll 12½ per cent then anneal at 1275 F for 5 hours.

Cold worked or annealed SAE 1030 steel exhibits an abrupt change in Charpy notched-bar impact properties at a given testing temperature. The highest testing temperature at which this abrupt change occurs, specifically, a change from a brittle to a tough type of failure, has been designated as the transition temperature. The following factors influence (elevate) the transition temperature of SAE 1030 steel in the Charpy notched-bar impact test; increase in grain size, increase in the degree of cold rolling from 0 to 20 per cent, stress-relief annealing after cold rolling. At a similar grain size the pearlitic SAE 1030 steel exhibits a lower transition temperature than the spheroidized steel in the Charpy notched-bar impact test. Charpy notched-bar toughness of quench-hardened and tempered SAE 1030 steel at low temperatures is superior to the toughness of the cold-worked pearlitic or spheroidized steel.

The assistance of Dr. M. A. Grossmann, Consultant to the Laboratory in the planning and conduct of the work is gratefully acknowledged. For permission to publish this article the authors are indebted to Mr. C. C. Fawcett, acting director of the Frankford Arsenal Laboratory, and to the War Department, Office, Chief of Ordnance.

Earth Mover Has Unusual Design Features

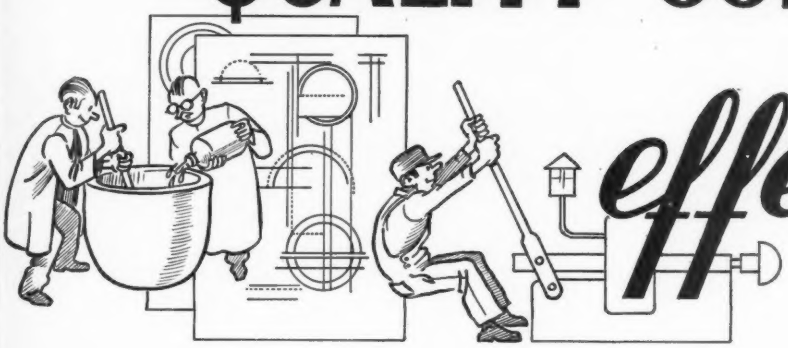
NEW approach to the problem of designing a machine for earth moving has been made in the development of the "Gradall". The machine manufactured by Warner and Swasey Co., has been so designed as to make possible six motions in addition to travel of prime mover.

Basis of the earth mover is a telescoping boom, the outer 12 feet of which is supported by and movable in the inner section, making the maximum length 24 feet, minimum 12 feet. Boom may be rotated 360 degrees in horizontal plane, 66 degrees in vertical plane, or 90 degrees on own axis. Grading or scooping tools placed at end of boom can be traveled in 116 degree vertical arc.

Powered by a 48-horsepower gasoline engine, a hydraulic pump delivers oil at 1000 psi to cylinders powering all motion.

Interesting features of machine are methods of boom support and table rotation. Inner boom is supported in outer by six hardened and ground steel rollers placed three at each end as seen in illustration. Boom is supported at inner end by annular ring rotating in frame on four steel rollers. Frame is held in large trunnions fastened to rotating table. Platform rotation is accomplished by roller chain engaging sprocket fixed to truck frame. At each end of a chain hydraulic cylinders rotate the platform, or hold it fixed in place. Boom extension and rotation as well as tool rotation are likewise accomplished by cylinders linked to members. The machine has been used for grading and contouring. Other operations include ditch or hole digging and hoisting.

Applying QUALITY CONTROL *effectively*



By T. D. Foy
Apparatus Manufacturing Division
General Electric Co.
Schenectady, N. Y.

QUALITY is not improved, costs reduced or losses eliminated by mere statistical analysis of production problems. A long-range approach which will assure successful quality control by providing adequate corrective action against variations which are the outgrowth of new designs, new materials, new processes, methods, machines, etc., is outlined by the author in this paper which was presented at the recent AIEE summer convention held in Detroit

DURING the war, the accelerated development of new designs, machines, methods, processes, etc., plus wider usage of standardization and interchangeable parts, gave rise to a need for, and wider utilization of, modern statistical methods of quality control.

Today industry is more "tolerance conscious" than ever before—it has to be. Limits gradually are being narrowed to approach the nominal dimension, increasing the precision with which parts must be manufactured and the need for exercising closer control over variations from that nominal. Mention of plus or minus 0.001-inch tolerances are no longer startling, for attention is now turned to tenths and hundredths of a mil.

As preparation is made to meet the demand for more and better products for more people at lower costs, there is increasing recognition of the need for an organized plan whereby a high level of quality can be established and maintained in manufacture.

In most manufacturing organizations use of quality control is begun on a small scale. A specific project that is particularly troublesome or costly is selected, the problem is analyzed using control charts, distribution curves, or sampling tables, corrective action indicated by the data is taken, and the results (improved quality, reduced losses, reduced costs, etc.) are tabulated and presented to management. With the approval of management, new projects for application are then determined, and the activity continues to expand and become more comprehensive as additional progress is made.

Gradually, however, it is recognized that the statistical method of controlling quality, though extremely useful in finding the cause of quality variations, is not a cure-all for manufacturing troubles. Quality is not improved, nor costs reduced, nor losses eliminated by mere statistical analysis of the problem, but rather *by corrective action taken on the basis of this analysis.* Direction of such action, however, requires well - organized engineering co-operation and co-ordination.

A logical decision as to the proper type of organization

for quality control hardly can be made without sufficient understanding of the problems involved in controlling quality. So in order to analyze the case for quality control completely, the various contributing factors will be outlined.

Factors Involved in Controlling Quality: What affects quality? Where is it affected? How is it affected? Generally, those items which affect quality can be classified as:

1. Men
2. Materials
3. Processes
4. Machines
5. Specifications
6. All others, such as dust, humidity, extreme temperature, etc.

More often than not, the reasons for quality variations cannot readily be allocated to just one of the items above; usually it is a combination of effects which are difficult to evaluate individually.

In a similar manner, those organizational functions in the performance of which quality is affected can be classified as:

1. Engineering
2. Material ordering
3. Planning and methods
4. Manufacturing
5. Inspection and test

Here again, a discussion of quality problems frequently involves more than one function of the organization before proper corrective action is obtained.

Quality Affected by Many Factors

When the possible combinations of what affects quality, where it is affected, and how it is affected are fully analyzed, it is found that, for the most part, any one cause of quality variations logically might be the responsibility of any or all functions of the organization, depending upon the type of quality problem involved. For instance, a problem involving the quality of material for a particular application might properly concern:

1. Engineering—from the viewpoint of original design, selection of material, tolerances, etc.
2. Material ordering—in selection of vendors, and type of quality guarantee requested
3. Planning and methods—in selection of machines, tools, processes, and design of jigs and fixtures
4. Factory personnel—from the viewpoint of the skill of operators, machine setup, employee morale, etc.
5. Inspection and test—from the viewpoint of judging quality of incoming material, processes, machine parts, sub-assemblies and the finished product.

To summarize, then, it can be seen that quality is affected by many different items, that several people exert varied influences upon quality, and that, to be fully effective, the organization must be so set up as to integrate the principles and co-ordinate the effects of the quality control program in all sections of a manufacturing division.

What is Meant by Quality Control: In the past, this phrase has been a source of considerable confusion, mainly because it has been interpreted in so many different ways. To some, it meant the usual overall co-operative efforts of engineers, planners, factory supervisors, inspectors, etc., to maintain a given standard of quality in product; to others, it meant the protective endeavors of inspectors and test men to assure quality products to the customers; to still others, it meant the organized examination of test and inspection data to be used as a basis for corrective action. More recently, it meant statistical analysis methods applied to engineering and manufacturing quality problems.

The attempt to clarify this new approach to the problem of controlling quality by use of the phrase "statistical quality control", has only tended to imply that the application of these methods is restricted to special instances requiring the services of a statistical expert. Such is not the case in good practice, however. A practical definition of this term, therefore, which includes the good points of both the old and the new statistical concept of quality control is "any organized management effort, statistical or otherwise, to establish and maintain required quality levels by suitable corrective and preventive action".

"more tolerance conscious"

Purpose of Quality Control Organization: To control the quality of the final product obviously necessitates controlling the quality of the product in *all* stages of the manufacturing cycle; from the conception of the engineering design through the final assembly. In the past, the factory often has been the recipient of, and unknowingly held responsible for, accumulated errors in engineering design, material ordering, planning and methods; errors which remained undiscovered until actual manufacture of the product began. Through the use of statistical quality control principles in new designs, machine and process-accuracy studies, sampling of incoming material and sub-assemblies, etc., much of this can be eliminated. Industry is demonstrating the effectiveness of the quality control program in controlling the quality of the product progressively.

The purpose of the quality control organization, then, is to: (1) Provide means for obtaining adequate data on current activity and progress and establish the means for taking immediate corrective action at the source of trouble and, (2) provide the means of educating all personnel affecting quality to the use of quality control principles so as to secure preventive action.

Needs Overall Organization

Type of Organization for Quality Control: With the background of the foregoing comments, the type of organization required can be presented. Consider first, locating the quality control activity in the inspection department. In justification of such a move, it might be pointed out that no one in the organization is called upon

to make more decisions regarding quality than the inspector; therefore, he knows more about what should be controlled than anyone else. Further, statistical sampling used almost exclusively by inspectors is a big part of the program to reduce manufacturing losses and inspection time. Again, it is the inspector who usually takes the data and makes the report concerning the quality of products and thus is in the best position to recommend what action is required to maintain quality.

However, quality must be built into a product; it cannot be *inspected* into it. Therefore, it is not believed that locating the responsibility for the quality control program in the inspection department can be fully effective. Inspection is most assuredly a part of the program; without inspection data there is little idea as to what progress is being made and where effort is to be concentrated, but the program is much bigger than the inspection department itself.

The inspector is in the best position to classify the type of poor-quality characteristics involved; that is, such off-standard items as defective castings, holes drilled off-center, oversized and undersized shafts, burrs, poor finish, etc. This is important, but far more important is determining the cause of the defective work and initiating the corrective action necessary to prevent recurrence of the

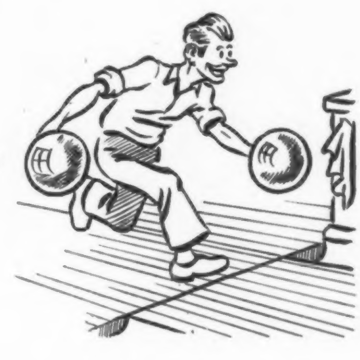
"organized examination"

same error in the future. This the inspector usually is not in a position to do, primarily because it is not considered a part of his job. An inspector, within the bounds of his present duties and responsibilities, can no more prevent the manufacture of poor-quality parts than the fire department can prevent fires. If quality is to be controlled, the tools for controlling it must be put into the hands of the people who are exercising the most influence over it. In this way, the job is done correctly the *first* time rather than the *next* time.

Consider next planning and methods. These most certainly influence the quality of the products in selection of processes, methods, machines, tools, jigs, fixtures, etc. But here again, although planning and methods should be a part of the quality control program, this activity is much bigger than the planning department itself. Similar arguments hold true to a certain degree in the case of engineering or manufacturing as possible locations for the sole responsibility of quality control activity.

Another type of organization which meets the need for such an important acivity can now be presented. There is frequent use of the expression "quality control is a management tool". If this is so, and generally it is agreed that it is, direction of the quality control program logically becomes the sole responsibility of management. Such a responsibility can be delegated, but can be effective only when it is fully understood by all concerned that the dele-

"requiring the services of a statistical expert"



gate represents management and, therefore, carries authority commensurate with his responsibility.

In a typical manufacturing division of General Electric, management is represented by an engineer responsible for the development and design of the product and a superintendent responsible for the progress of the product after the design has been completed and released for production. Those persons who are to direct the quality control program must, logically, report to these representatives of management; a quality control co-ordinator in charge of the activity reporting to the superintendent, and a quality control engineer working with this co-ordinator reporting to the engineer in charge.

With this arrangement, engineering activity under the direction of the quality control engineer, and material ordering, planning and methods, manufacturing, inspection and test activities under the direction of the quality control co-ordinator can be organized to produce most effective results. In no way is this program to remove or replace the responsibility of the foreman for controlling the quality of the work under his direction, nor the planner and the engineer of their responsibility in their quality efforts. Instead, it provides them with a method of doing

"authority commensurate with responsibility"

a better quality-controlled job through the application of distribution curves, control charts, and sampling tables to engineering and manufacturing quality problems.

Large, modern organizations of today are highly com-

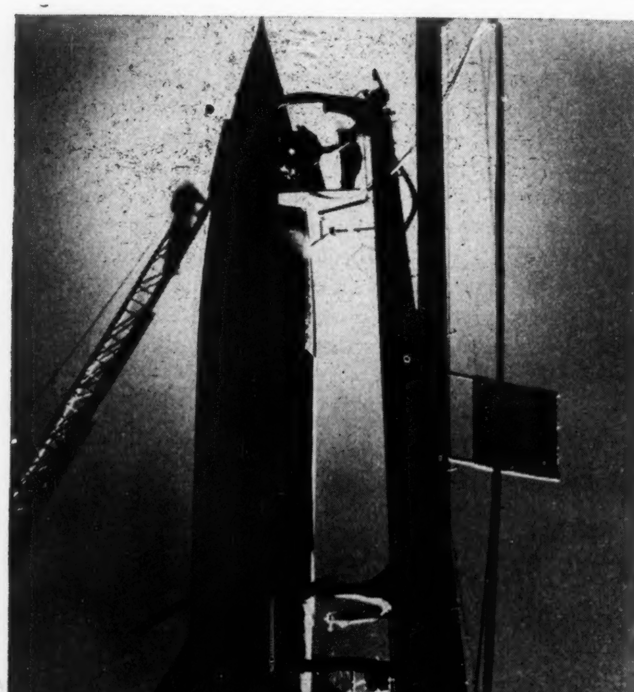
plex, intricately functionalized mechanisms with many divisions of responsibility, each function making a special contribution to the ultimate goal of building a high-quality product at low cost which will provide service according to

"no protection against rise of future problems"

customers' requirements. It is in contending with this problem that the quality control co-ordinator can make his greatest contribution. Not only is he concerned that the causes of quality variation be determined but also that corrective action be taken immediately by persons responsible. As a representative of management, removed from the influence of functionalized barriers, the quality control co-ordinator can best organize and propel the co-operative efforts of those sections most vitally concerned with the quality problem.

One point which again should be emphasized is that quality control activity must be organized to consider two approaches. The short-range program involving the attack on specific projects is productive of immediate results which can be evaluated in terms of quality improvement or cost reduction over some previously-established level. However, this approach is more or less a temporary expediency designed to overcome undesirable quality variations in products under immediate consideration. It usually does not provide adequate protection against the rise of future problems which are the outgrowth of new designs, new materials, new processes, methods, machines, etc. If each of these functions can influence the quality of a product, then each must be provided with the method of preventing undesirable variations in quality. This is the long-range program responsibility of the quality control co-ordinator and the quality control engineer; to so integrate the principles of quality control that it becomes a tool of everyday usage for doing the job right the first time. To be fully successful, such a program requires full management support, proper organization, wholehearted, overall co-operation.

Seek Super V-2 Missile



ROCKETS that will surpass the German V-2 in speed, distance, accuracy, and effectiveness are the primary objectives of a program of guided missile research being conducted jointly by General Electric and the Army Ordnance Department.

This research program was started in November, 1944, with the study of captured V-2 rockets in Europe. Special facilities have since been provided by the Army at Schenectady for testing new types of rockets and motors.

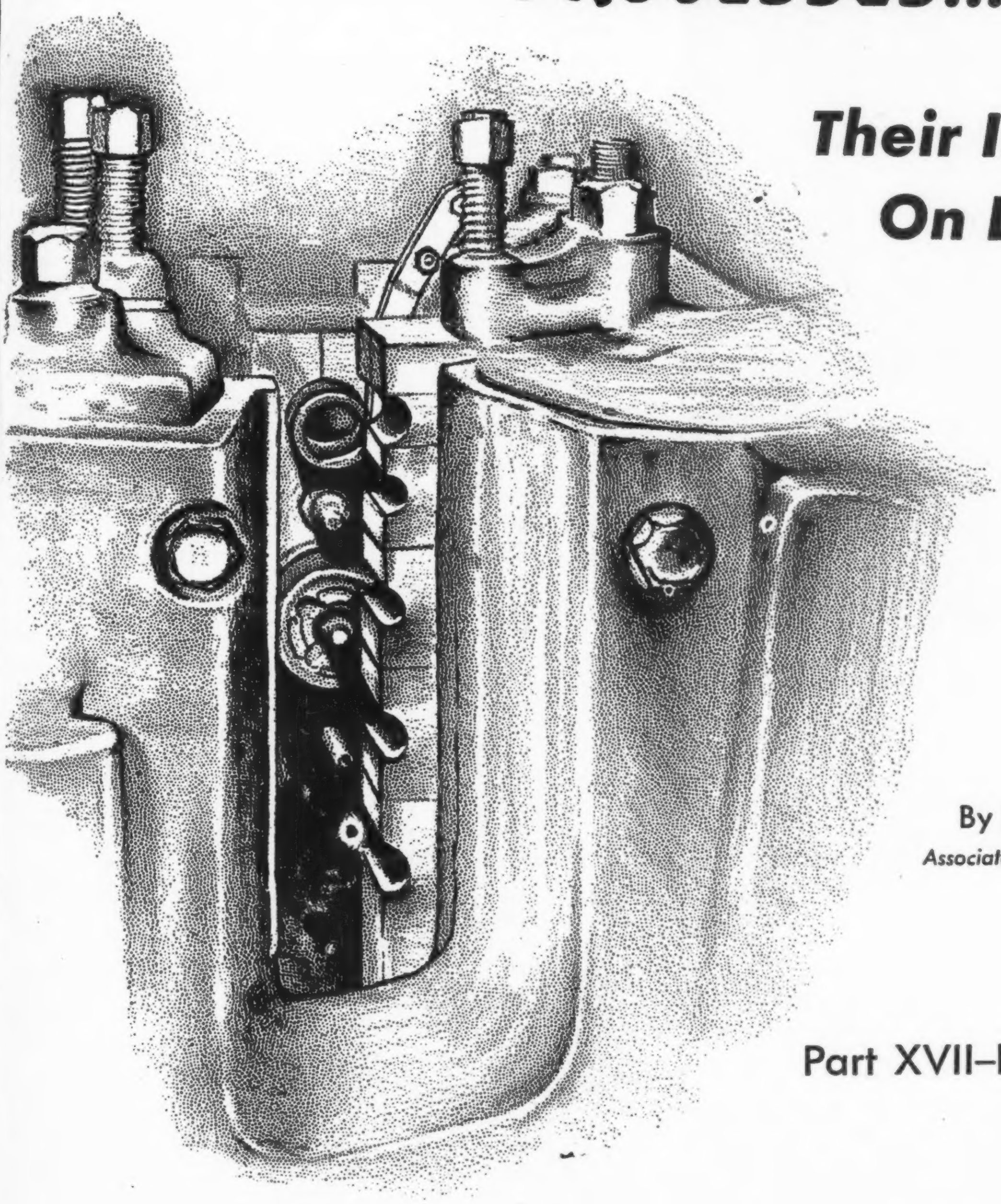
Propulsion research for missiles involves experimentation with rocket engines, ram-jets, and combinations of both. The ram-jet is, of course, simply a cylinder which compresses the air through speed in flight, feeds and ignites fuel in the combustion chamber, and derives forward motion from the thrust produced by expanding gases which blast through the nozzle. Thus, being dependent upon oxygen from the air compressed, ram-jets are subject to definite range-altitude limitations in that their ceiling cannot greatly exceed that of aircraft. Hence the concentration upon the possibilities of the rocket and rocket combinations.

Development and use of new, more powerful fuels for these units creates many metallurgical problems. Materials utilized not only must have great strength, but must withstand the corrosive actions of these new fuels, and resist any melting incident to the tremendous frictional heat generated in flight.

Among the flight problems which remain to be solved are those of increased gravitational effects on long, flat trajectory, and possible disintegration on the return to earth with a high-angle trajectory.

PRODUCTION PROCESSES...

Their Influence On Design



By Roger W. Bolz
Associate Editor, Machine Design

Part XVII–Hot Upsetting

ASSOCIATED with and often used as an alternative to the cold heading process discussed in Part XIII (M.D. July, 1946) is that known variously as hot upsetting, hot heading or machine forging. The original hot-heading machines, like those used for cold heading, were developed primarily for the production of headed fasteners of various kinds. Today, however, fasteners are but one of a tremendous variety of machine parts which are manufactured rapidly and economically by means of the hot upsetting method.

In machine upsetting hot metal, the material—usually "cut-to-size" rounds rather than wire—is forged by dies moving along its axis in the manner previously described in the article on cold heading. Except on the smaller size material handled, straight lengths rather than coils are used and, in a great many cases, parts semifinished by preliminary processing methods are completed in the upsetting machine, *Fig.* 1. Some scaling often is encountered with heating unless an induction method is used and consequently most parts have a finish somewhat inferior to that resulting from cold heading. Hot upsetting, however, offers a greatly increased field of design variations since hollow, bent and complicated formed parts can be produced easily from any material suitable for forging, *Fig.* 2. Much greater volumes of metal can be upset hot than cold and, in addition, any design that can

Fig. 1—Above—Hamilton 25S-T aluminum-alloy propeller blade showing heavy root completed on the upsetter

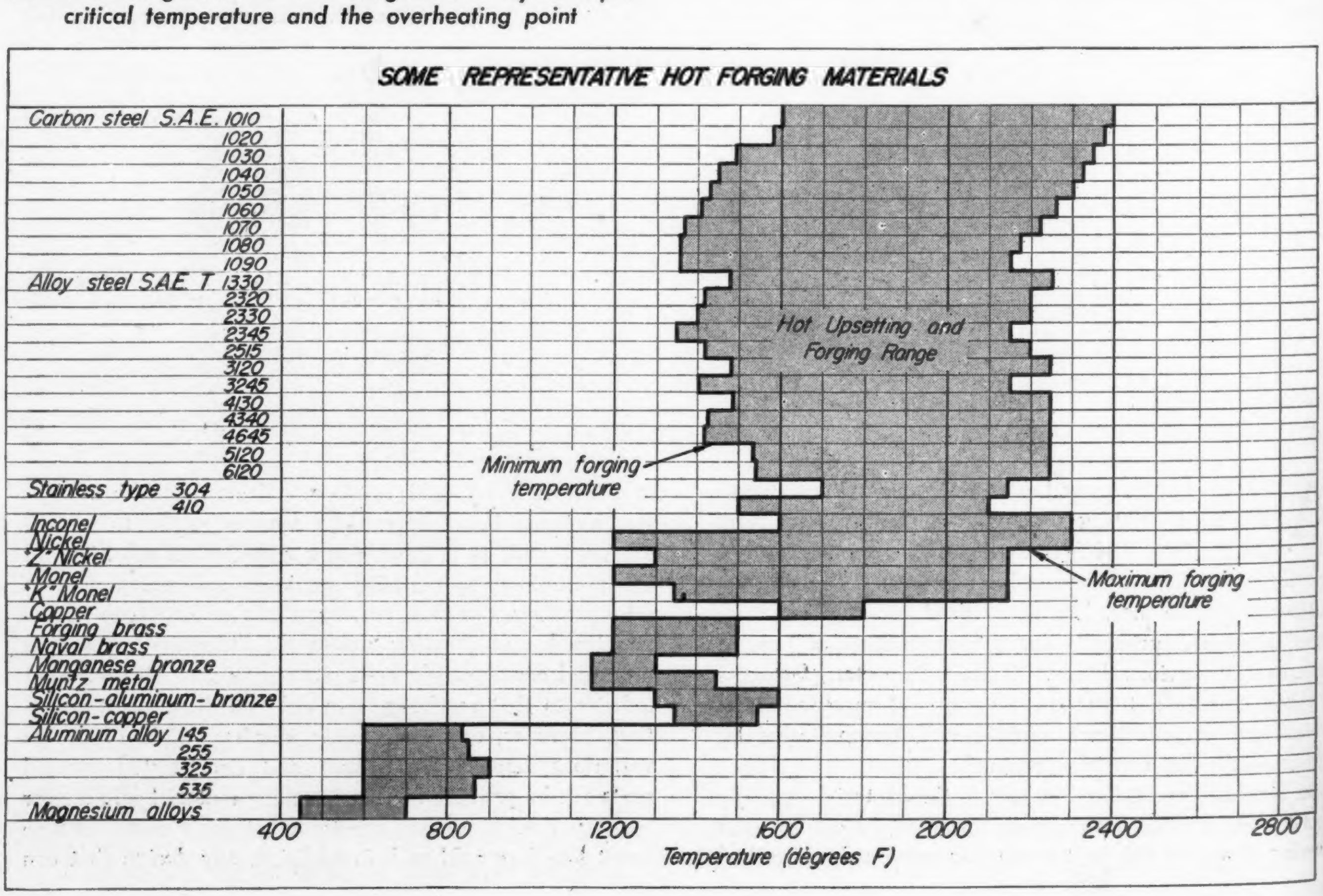
Fig. 2—Below—Metals possess an almost infinite capacity for hot working within a finite range limited by the uper critical temperature and the overheating point

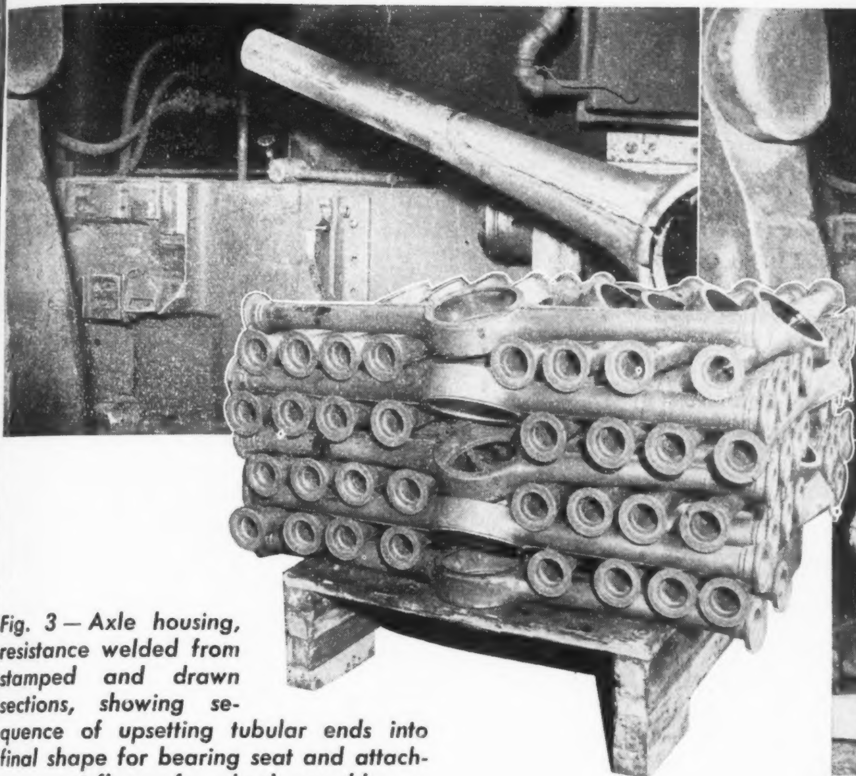
be produced cold can also be made by the hot process but the reverse is not true.

Inasmuch as most metals have a definitely limited capacity for cold working when subjected to pressure beyond the yield point and below the ultimate strength, severe limits on design and size naturally are present. Although larger wire sizes can be worked, as previously noted, standard machines for cold heading are limited in capacity to a maximum of one-inch diameter. Mechanical restrictions, therefore, in addition to the fact that workability or plasticity of various metals increases as temperature increases, *Fig.* 2, makes it customary to turn to hot upsetting for mass producing many types of parts from stock 7/8-inch diameter and over.

In forging metals, either an impact hammer blow or a squeeze pressure is used. Hot upsetting forging machines, like cold headers, are essentially double-acting horizontal presses, although as a rule not automatic in operation except on certain long-run production jobs. Where high production is desirable on a standardized part, straight bar stock for both large or small parts is fed through special furnaces to the machine. In some cases with small parts, stock in coil form with special continuous heating furnaces can be used. However, induction heating methods are rapidly coming into use and promise both increased production and better products, as precisely controlled heating easily is achieved and steel is nearly scale free.

After heating bar or billet stock for the machine upsetter, it is fed to a stop in the stationary half of a split grip die either automatically or by hand and, when the cycle is actuated, the moving half of the grip die closes against the stock, holding it while the heading ram forces





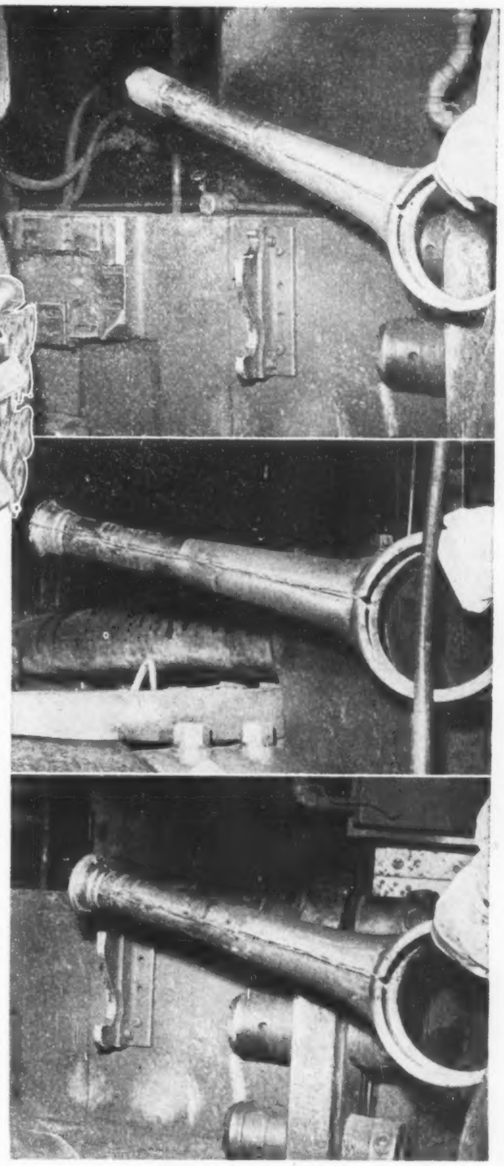
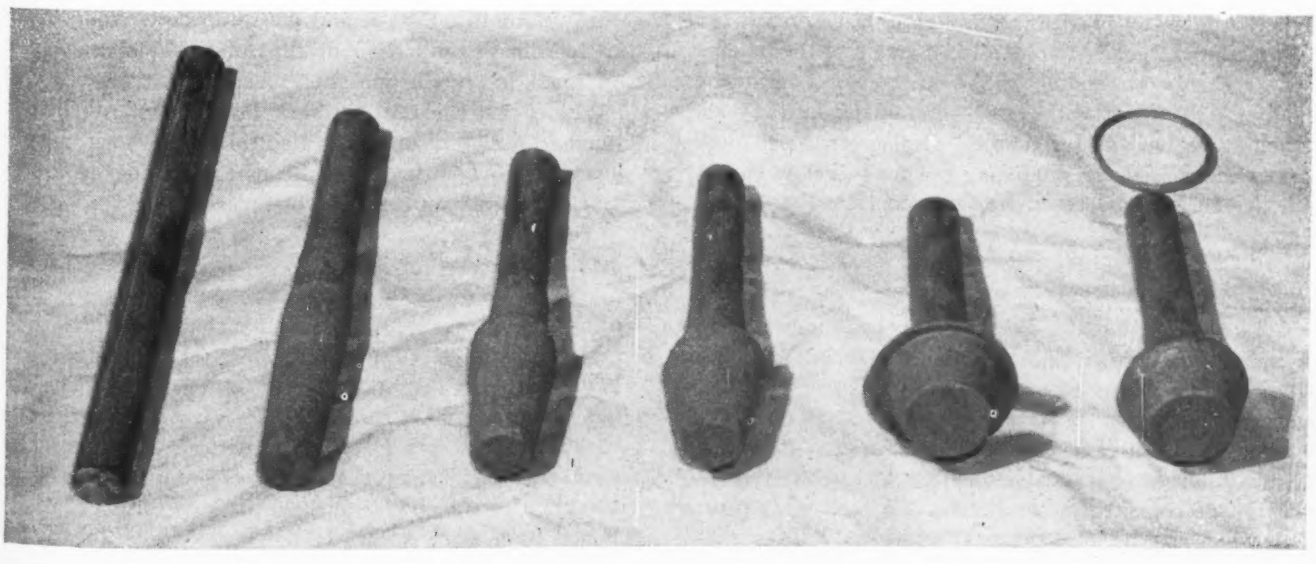
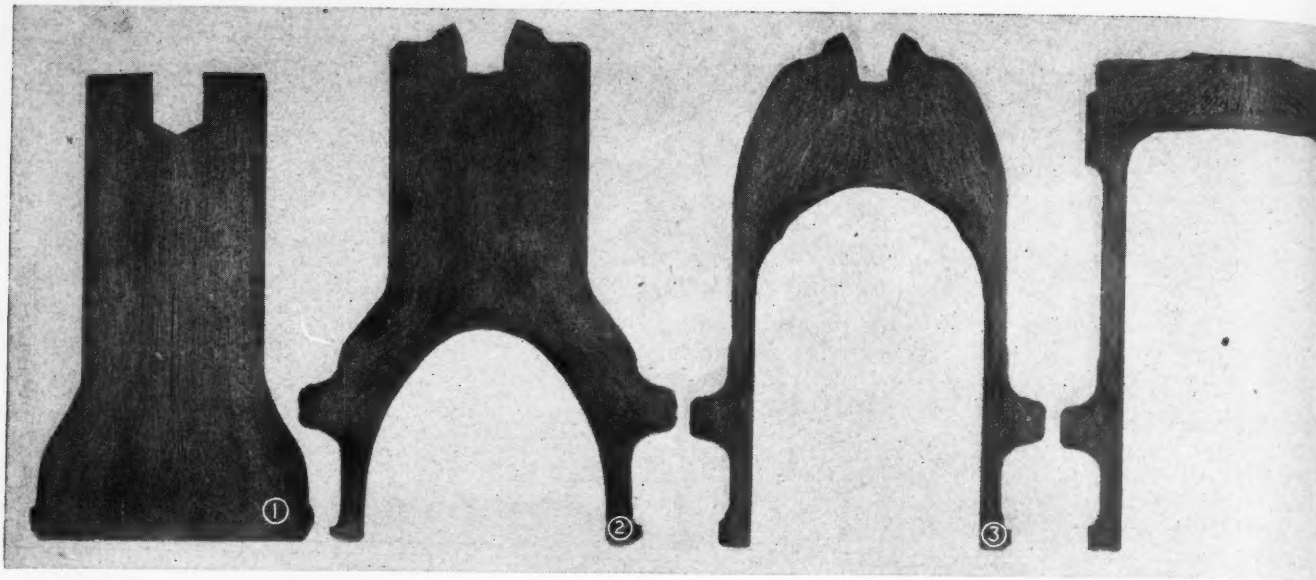


Fig. 3 — Axle housing, resistance welded from stamped and drawn sections, showing sequence of upsetting tubular ends into final shape for bearing seat and attachment flange for wheel assembly

the exposed section of heated material into the die cavities. Although many upsets can be made in one automatic operation, most designs require movement or working of a large amount of metal and consequently the majority of parts are produced in a number of separate steps. These usually are set up in a composite die arrangement and the part moved progressively from the initial step to completion without reheating, *Figs.* 3 and 4. Limiting factors which largely determine the number of upsets necessary to shape the desired volume of materials are: (1) Maximum length of unsupported bar that can be upset in one blow without buckling is three times the bar diameter; (2) lengths or volumes greater than three diameters can be upset providing that the diameter of upset (i. e. die cavity) is limited to a maximum of 1.5 times the bar diameter; (3) where the length of upset or volume is greater than 3 diameters the amount of unsupported stock extending beyond the die cavity cannot exceed one diameter; and (4) wall thickness of tubing cannot be increased externally more than 25 per cent at a blow, al-

Fig. 4—Below—Pinion gear upset forging showing sequence from 1½-inch diameter rough stock through five stages to the final piece ready for machining





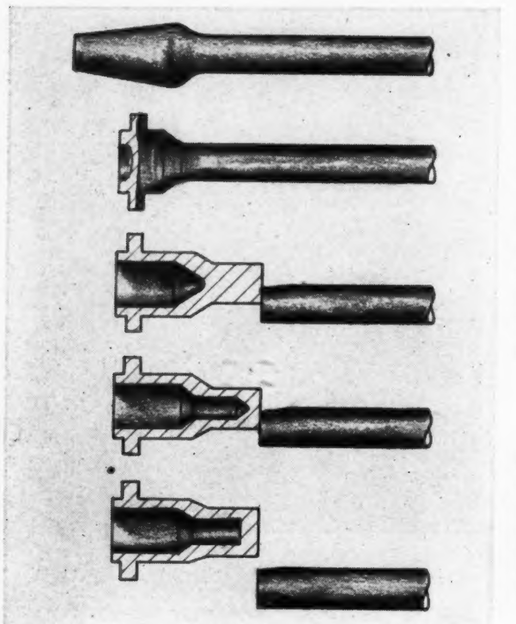


Fig. 5 — Above—Sequence used in machine upsetting steel marine engine cylinder barrels. Replacing conventional forging methods, these barrels were machine upset in four passes in one heat, reducing by almost 60 per cent, the previous rough forging weight. Hole in end is for handling the 4¾-inch diameter by 17¼-inch long billet

Fig. 6 — Left — Progressive steps in deep piercing a representative upset part. Both outside diameter and bore can be stepped. The flange simplifies the piercing operation

though internal upsets are almost unlimited since the arch effect prevents internal buckling.

In addition to plain upsetting, the transverse action of the gripper die and the longitudal action of the punches can be utilized separately or simultaneously for swaging, bending, shearing, extruding, trimming, slotting, and punching as well as for internal displacement—an operation widely used for producing hollow socket wrenches, cylinder barrels, and other deep-pierced jobs, *Fig.* 5.

Hollow Designs Advantageous

Care in designing the dies and punches for each stage of upset results in optimum grain refinement and flow. Study of four stages in *Fig.* 5 reveals this action. Internal working action provided by punches makes it advantageous to utilize hollow designs rather than solid in many cases. Permissible heating temperature of the material of course, *Fig.* 2, dictates the high temperature limit which can be used for maximum ease in forging and piercing. Ability of the piercing tools to withstand the tremendous pressures developed when working in the low temperature range severely limits the practicability of low-temperature piercing in high production.

Deep piercing of a representative part is shown in *Fig.* 6. The extrusion principle is not used in this operation because wear on punches and dies makes it practical only for nonferrous materials such as aluminum, lead or brass. All straight-carbon and alloy steels, therefore, as well as all forgeable nonferrous metals are handled ty piercing rather than extrusion.

Commercial hot-upsetting machines now available will handle work the stock for which ranges from around 3/4-inch to 9 inches in diameter. Parts of almost any length or shape can be handled, *Fig.* 3, providing the upsetting rules are not violated.

As with cold heading, production quantities in hot upsetting must be sufficiently great to offset the cost of the dies. Naturally, the die and die setup costs would reach an inordinate proportion of the total cost of each part on small lot production, but where quantities are well into the thousands it becomes insignificant. Where a standard design of upset can be utilized, however, regardless of the overall length of the part, quantities as low as 100 pieces and up can be economically produced.

Thus for machine upsetting in a small-size range just overlapping that of the larger cold-heading machines, quantities normally duplicate those found economical for cold-heading practice, especially for similar designs. However, economical production rates for complicated or large parts may drop to a few pieces per hour. The cylinder barrels shown in *Fig.* 5 were produced at a rate ranging from 38 to 50 per hour on an 8-inch machine with only a nominal stock allowance for final finishing operations. Some 6000 to 8000 of these forgings are obtained from a set of dies before resinking of the cavities

is necessary and several resinkings can be made before new dies are required. Die life, of course, depends greatly upon the forgeability rating of the material used. Ratings listed in Table I of "Part IV—Die Forging" of this series gives a good indication of comparative characteristics. As a general rule, piercing punches have a life of from 8000 to 12,000 pieces per redressing.

A highly interesting adjunct to the upsetting process is that known as "gathering." Developed specifically for speeding the production of extremely large-diameter flanges such as those necessary for diesel engine valves, the so-called "electric-gathering machine," *Fig.* 7, gathers stock into cylindrical or spherical shapes for subsequent upsetting. By this method the required volume of material is gathered into a satisfactory shape and passed directly to the upsetter where one blow finishes the part, *Fig.* 8. Thus, die design is simplified and furnace heating eliminated.

Essentially, the gathering machine consists of a lower electrical contact of high-strength and heat resistance, a crosshead with scissor-type electrical contacts, means for adjusting the initial gap between upper and lower contacts, provision for automatically raising the cross-slide at a predetermined rate, and a source of power at low voltages and high current densities. The initial gap, the ratio of cross-slide travel to ram travel, the hydraulic pressure, and electrical power settings depend, of course, on the material, diameter and length to be gathered. By suitable selection of these factors, material may be gathered as desired at any point along a part, using round, square or hexagonal material from 3/16 to 2½ inches in diameter or across flats, the main requirement being hot workability and electrical conductivity.

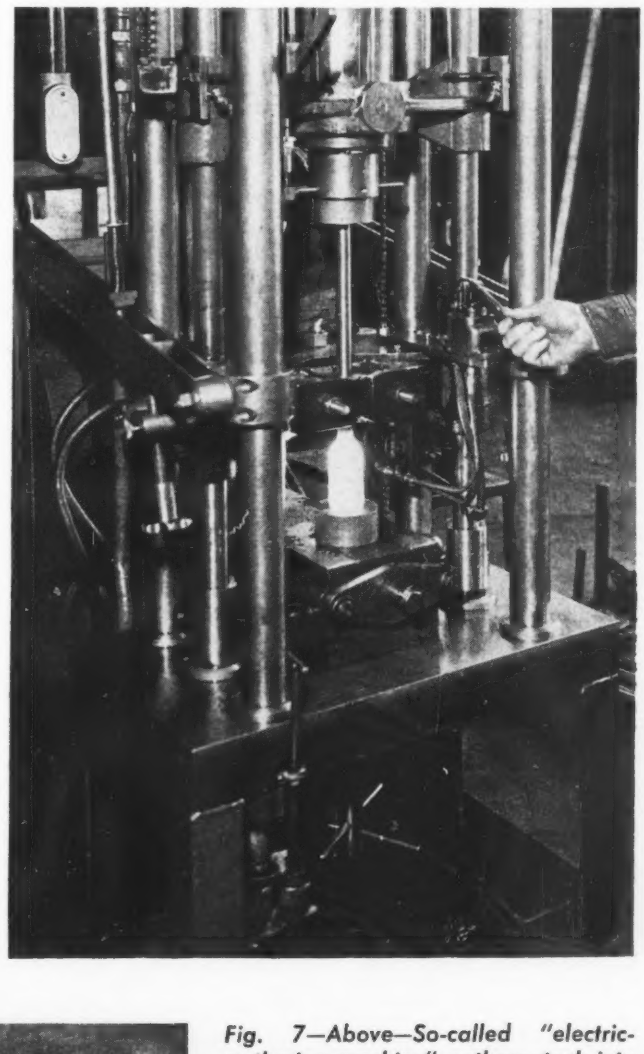
Gathers of 30 Diameters Practical

Although with regular machine forging the limitations on length or volume of upset somewhat restrict the design of parts, in a few special cases, the gathering of as much as, say, 2½ inches of ⅞-inch stock has been accomplished with special sliding dies. However, with the electric machine, gathers up to 30 diameters easily can be made in production and indications are that this limit can be exceeded. Thus the cost of complicated dies as well as controlled furnace heating can be reduced to a point where even small quantities of unusual upsets are practical.

Tubing also can be gathered in this way for upsetting. The critical factor here, however, is wall thickness rather than diameter or D/t ratio. Stainless steels can be processed down to 0.065-inch wall but scrap is high and for maximum economy it is not advisable to use less than a 0.095- inch wall. On low-alloy tubing, 0.250-inch wall is about the practical minimum since this material has a greater tendency than stainless to develop folds during processing.

Not a great deal unlike the electric gathering machine, the standard flash-welding machine is fast finding use as a production automatic upsetter of rod and tubing specialties. By this method blank shapes for cams, flanges, gears, and other parts to be made integral with a shaft can be easily and cheaply produced with all the assets of an upset forging.

DESIGN: A part suitable for hot upsetting, much as



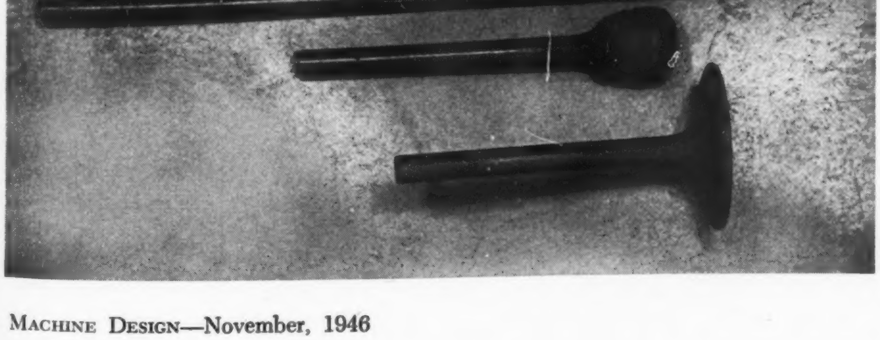


Fig. 7—Above—So-called "electric-gathering machine" gathers stock into cylindrical or spherical shapes which can be directly upset into finished or semifinished parts

Fig. 8—Left—Large-volume upsets are electric-gathered and finish upset in two simple operations

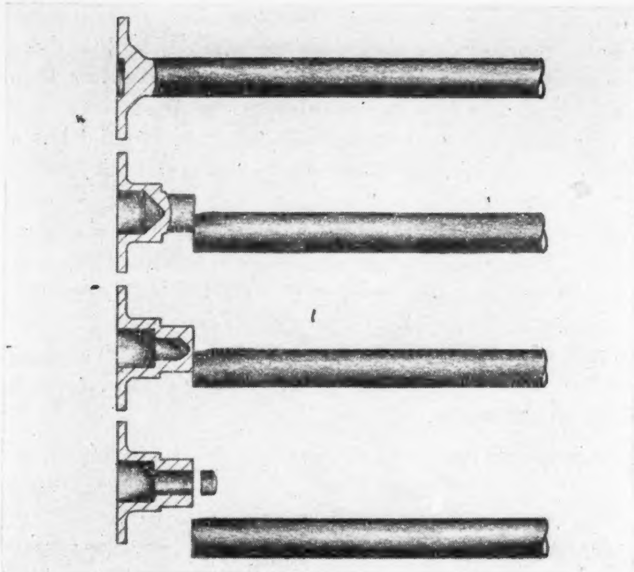


Fig. 9—Flanged hub with stepped bore pierced through. The forging is sheared from the bar at the same time the small bottom slug is punched out by the last ram tool

with those for cold head-heading, would require an excess of material, machining or forming if produced by other methods. Unlike cold-headed designs, though, complexity and rearrangement of material can be great, taxing to the limit the hot workability of the metal used. Although uniform circular cross sections are most easily produced, practically any variety of cross section can be had—hexagonal, square, conical, oval, hollow, etc. It is practical to upset not only a flange on the extreme end of a part but also one or more at intermediate points along the part.

Designs which can be held within the three-diameter volume maximum outlined in the preceding limitations, of course, can be produced most easily; often with only a single blow. However, where maximum upset is desired to produce an unusual design, die cavities can be held to a diameter of 1.3 times the stock diameter and extremely large flanged portions can be produced although as many as seven separate blows may be necessary. Naturally, the more cavities needed in a die, the greater is the cost and slower the production. Consultation with a reputable producer will assure a practical design as well as assist in obtaining simplification.

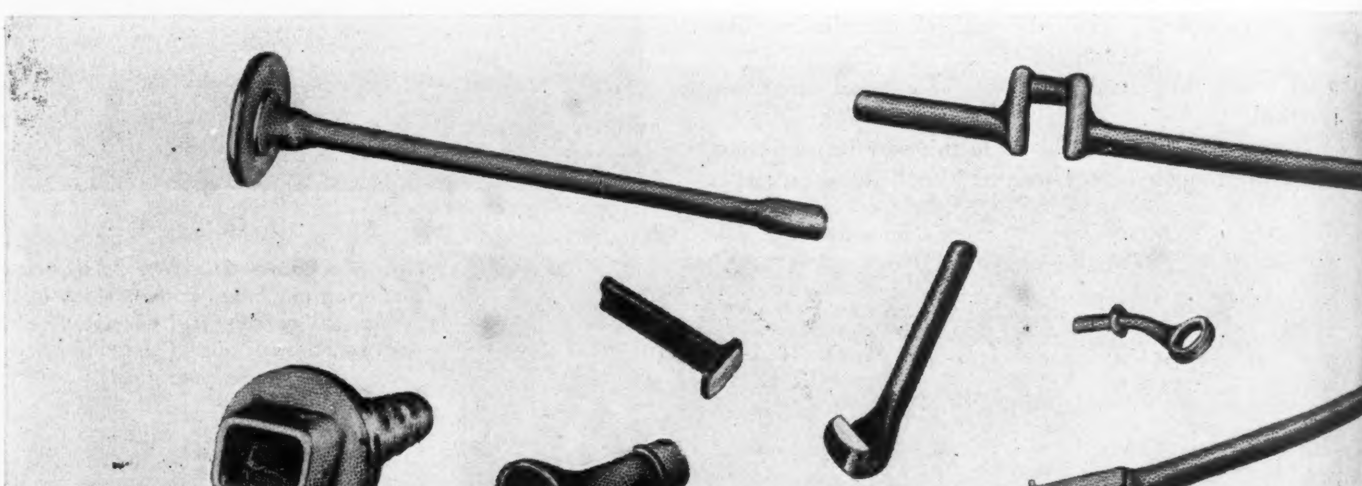
Inasmuch as split grip dies are used, tapers can be of any variety. Sharp corners should not be specified unless finish machining operations planned can produce them. Generous radii or tapers at corners or stepped junctions will help to increase useful die and punch life considerably.

Flat-Bottom Bores Practical

Hollow parts can have holes of practically any size pierced in the process and the pierced portion can be perfectly straight in bore or stepped, *Fig.* 6. By the method shown in *Fig.* 6 flat sheared ends can be had and, if desired, pierced parts may have flat-bottom bores, the only restrictions being that the square-ended bottoming tool must have rounded corners.

Bores on pieced parts also may be carried through by a punch which operates simultaneously with the shear. *Fig.* 9. Scrap loss in the form of a slug, as readily can be seen from *Fig.* 9, is insignificant. Depth of pierce holes is restricted somewhat owing to the fact that at a single piercing the length of unsupported stock ahead of the piercer (between the upset position and the point where the stock is gripped) cannot exceed three times the diameter of the bar. Slender steel sleeve forgings having pierced bores some 7.5 times their diameters are reported successfully produced at 200 per hour with a punch life of 8000 to 10,000 pieces per dressing.

A variety of unusual designs produced on a modern upsetter are illustrated in *Fig.* 10. From these an idea of the tremendous field of design applications readily can be gained. In each case a very minimum of machining is necessary to complete the part. Accuracy in detail obtainable necessitates the allowance of only about 1/16 to 3/32-inch for those sections requiring finish machining.



MATERIALS: Of the common engineering materials available, any one suitable for forging, as outlined in "Part IV—Die Forging" of this series, is equally suitable for upsetting. Many metals unsuitable for cold forging are easily handled hot in production. Although hot forgings generally are more costly than cold, more intricate designs can be produced and, in certain cases such as with silicon-aluminum-bronze, stronger parts result having few or no dangerous internal mechanical stresses. Forgeability of various metals are compared with the base SAE 1020 in TABLE I of Part IV.

From the chart in *Fig.* 2 which shows the forging range of common metals, it readily can be seen that the temperature range through which many of the nonferrous alloys flow easily is relatively narrow. Danger of overheating with these metals is correspondingly high and control must be precise. The relatively high forging temperature of copper coupled with its tendency to form hard black oxide, which has serious erosive action on dies, makes it far more suited to cold heading wherever possible.

Normalizing Improves Machinability

For maximum refinement of grain with steel parts the finishing temperature in hot working should be just above the critical range to assure only time for the austenitic grains to assume a fine and uniform normal structure. However, as this is seldom possible on fast production where die life depends upon working with the material in highly plastic condition, it is advisable to normalize all parts after forging to obtain maximum grain refinement. In addition, normalizing improves machinability and relieves internal cooling stresses.

Although the electric gathering process has been used primarily for processing carbon and alloy steels, other materials such as brass, bronze, aluminum, magnesium, Monel, stainless, etc., also can be upset. Without doubt the process can be utilized to advantage on any material which can be hot worked satisfactorily and is a conductor of electricity.

TOLERANCES: As a rule, dimensional limits held in cold heading are closer than those held in hot upsetting. However, actual tolerances in most cases depend upon the finish machining operations involved. It goes without question that wide tolerances increase practical die life, in direct ratio, and costs in inverse ratio.

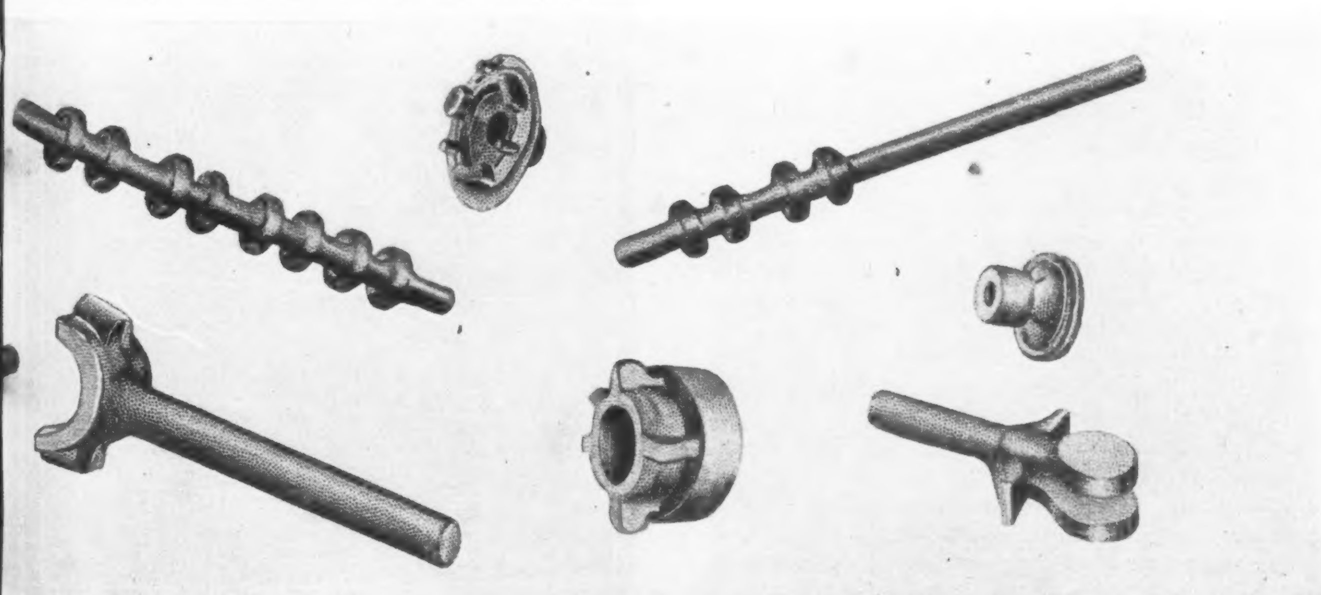
Thickness tolerances listed in TABLE II of Part IV, when pertaining to upset forgings, apply to those dimensions in a direction parallel to the travel of the ram of portions of a part that are enclosed and formed by the die cavities. Width and length tolerances, composed of shrinkage and die wear and mismatching tolerances, shown in TABLES III and IV, apply to the width or length in a direction perpendicular to the direction of travel of the upsetting ram.

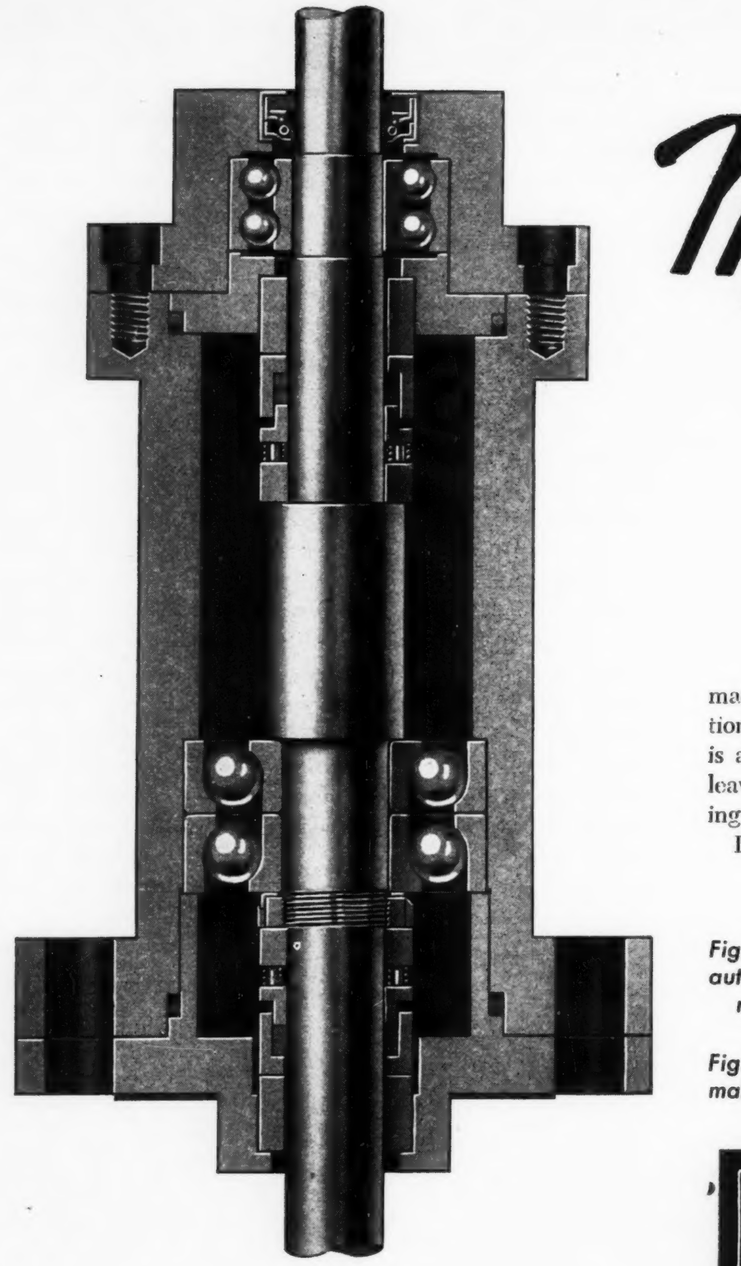
In machine forging work, as a rule, some draft should be allowed on portions parallel to the ram axis to minimize die and punch wear. However, parts with no draft can be produced when desirable to eliminate machining operations. Generally, from one to 4 degrees of draft are specified depending upon the material action and design. Standards adopted by the Drop Forging Association call for a nominal draft of 3 degrees for outside surfaces with a commercial maximum of 5 and a close limit of 4. On the inside of holes and depressions the nominal draft angle is 5 degrees while the commercial maximum is 8 and close limit, 7. Fillet and corner radii tolerances are the same as those for drop forgings.

Collaboration of the following organizations in the preparation of this article is acknowledged with much appreciation:

Aluminum Company of America (*Fig.* 1) .... Pittsburgh
Bridgeport Brass Co. .... Bridgeport, Conn.
Chrysler Corp. (*Fig.* 3) .... Detroit
Drop Forging Association .... Cleveland
Ford Motor Co. (*Fig.* 4) .... Detroit
Henry Ford Trade School .... Detroit
International Nickel Co. Inc. .... New York
Lamson & Sessions Co. .... Cleveland
National Machinery Co. (*Figs.* 6 and 9) ...... Tiffin, Ohio
Standard Tube Co. .... Detroit
Tube Turns Inc. (*Fig.* 5) .... Louisville, Ky.
Thompson Products, Inc. (*Figs.* 7 and 8) .... Bell, Calif.

Fig. 10—Below—A few good examples of production upset parts help to visualize the wide field of application





Mechanica

By Douglas R. Lewis
Mechanical Engineer
Newark, N. J.

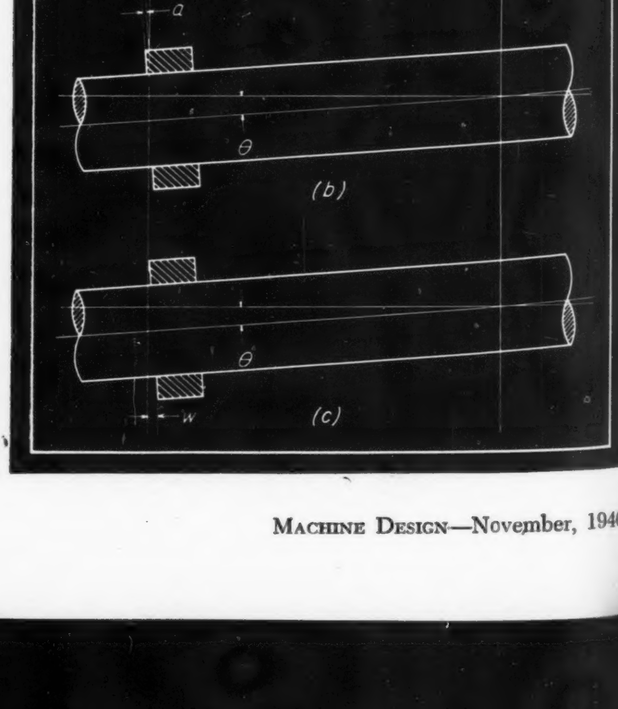
IT CAN be said with considerable assurance that a machine designed to accommodate conventional type packing glands is rarely if ever well suited to mechanical seal application. In many instances this is unfortunate, because the machine manufacturer is reluctant to change his patterns or general design unless he is certain that the seal will be successful, and the seal manufacturer cannot assure satisfactory seal performance when the machine has inherent weaknesses from the standpoint of good sealing. The length of time that the resulting stalemate continues depends on the seriousness of the stuffing box problem, the pressure applied by the machine users for improvement, and competition.

If these forces eventually compel a change, the designer should try to avoid any compromise with a packed box. Nothing short of the ideal design for mechanical seals should be his goal. If it is necessary that the user have a machine in which he may use packing or seals at his option, it will be found in general that what is good for seals is also good for packing, but an adequate packing design leaves much to be desired so far as good mechanical sealing is concerned.

In cases where machines are being designed for the first

Fig. 18—Bearing assembly built around double seal for autoclave or pressure vessels provides necessary shaft rigidity for sealing under a wide range of conditions

Fig. 19—Position of seal for shaft deflection, showing normal conditions at (a), when shaft deflects without axial movement at (b) and with axial motion at (c)



Seals

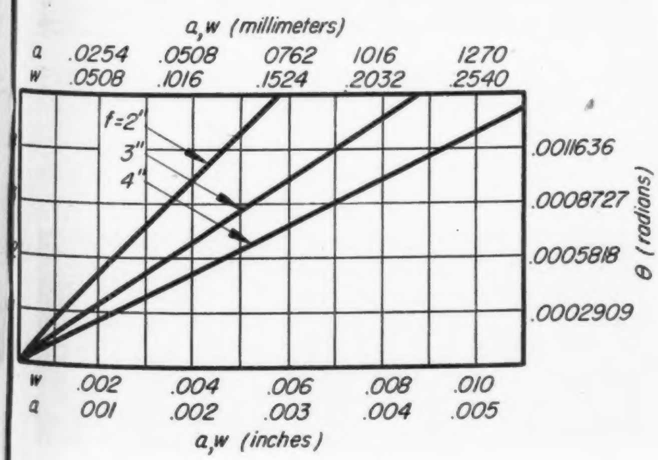
Part IV—Application Factors

time, and seals are to be used, the engineer is not so handicapped with an inventory of drawings, patterns, jigs, special machinery, and tradition. The advantage is indicated by the almost universal success of seals where they have been applied to machines "built around them".

An excellent example of this is shown in *Fig.* 18, an autoclave bearing assembly made in a complete unit, sealed and ready to be mounted on the autoclave. It is shown with a double seal but is made to accommodate any type of seal, and is sufficiently flexible in design that it can be used with practically any combination of speed, pressure, temperature, shaft diameter, shaft overhang, radial load, and thrust load. It can be supplied with an inverted flinger and drainage compartment which prevents any contamination whatever. The unit marks a distinct advance in autoclave technique, but the principles involved are fundamental and applicable elsewhere.

These principles are three in number, and the machine designer who contemplates the use of mechanical seals should keep them in mind at every step. First, the life and efficiency of a mechanical seal are directly proportional to shaft rigidity. Second, means should be provided for fastening the stationary seal member to the machine so that its face remains flat and normal to the axis of rotation except in cases where the stationary member is truly self-aligning. Third, bearings and locknuts, flanges, handholes, spacers, threaded pipe connections, couplings, and other machine elements which confine the seal or limit accessibility to it should be disposed to allow maximum ease of seal installation, inspection, and maintenance.

Fig. 20—Axial motion and leak opening plotted against deflection angle for three seal outside diameters



Shaft Rigidity

Lack of rigidity in a shaft, affecting seal operation, can be of three types—deflection, whip and end play. The first is not harmful to good operation of seals which have self-aligning stationary members, or to seals which have fixed stationary members and self-aligning rotary members if the angular velocity is low enough to permit full flexure and recovery every revolution. The second is harmful in all cases except at extremely low speeds. Whether the third is harmful or not depends on the rapidity and magnitude of the end play.

Deflection would never give trouble if it were steady, but it seldom remains so. A typical example is found in a pump feeding liquid into a tower or column, the shaft of which has the normal position shown in *Fig.* 19*a*. As the liquid level rises the moments acting on the shaft change. The deflection angle varies from the point where the level is lowest and the most power is being absorbed, through the point where the pump is operating at the most efficient point on its curve, to the point where the tower is full. As the pump approaches shut-off, the deflection angle increases. During these changes the axis of rotation moves through some angle θ, and, unless the stationary seal member can align itself to each new position of the rotating member, the relation between the two faces approaches the condition shown in *Fig.* 19*b*.

Materials used in seal faces are selected primarily for their wear resistance. Considering the duration of the misalignment and assuming that the rotating member is free

Fig. 21—Stationary seal face scored due to warping of flange having two diametrically opposite bolts

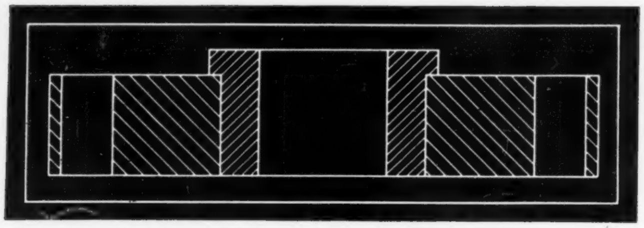
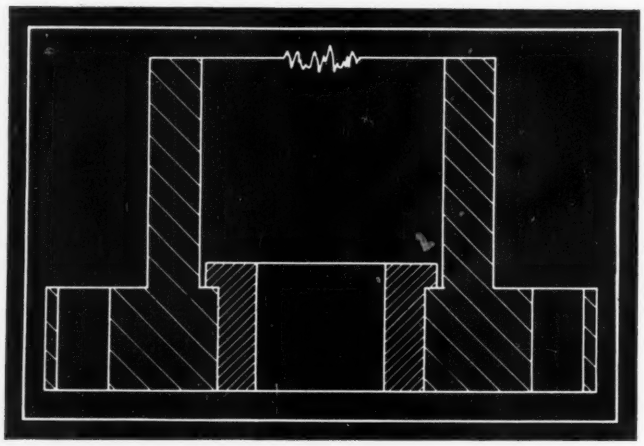


Fig. 22—Above—Shear strength of shoulder limits assembling pressure of a press fit

Fig. 23—Below—Relation of bushing shoulder to housing bore not visible in this design



to move a small amount axially, it is unlikely that the rotating face will wear into the stationary one the distance a in *Fig.* 19*b*. It is more probable that the rotating member will move axially the distance a away from the position it would occupy on the deflected shaft if the stationary face were not there. This occurs even at moderate speeds, and produces a gap for leakage on one side of the shaft, w in *Fig.* 19*c*.

The value of the quantities a and w depend in part on the forces holding the faces together, the hardness of the face materials, the speed, and the elasticity of the joint between the shaft and the rotating member in a plane normal to the shaft. If it is assumed that the forces holding the faces together are not great enough to cause brinelling of a pair of faces of given hardness, and that the relationship between speed and joint elasticity allows no self-alignment, then

$$a = \frac{f}{2} \tan\theta - \left(\frac{\cos\ \theta}{g} - g\right) \qquad (1)$$

where a = distance rotating member moves axially, f = rotating seal face diameter, θ = deflection angle, and g = distance from seal face to intersection of normal and deflected axes of rotation.

Also,

$$w = f \sin\theta \qquad (2)$$

where w = width of opening at seal face.

It is interesting to note that the distance from the faces to the point where the normal axis of rotation and the deflected one intersect does not enter into the second calculation, and affects the first one only by the amount $(g/\cos\ \theta) - g$. Since the cosine of very small angles is practically unity, the quantity $(g/\cos\ \theta) - g$ is negligible. Assuming a maximum deflection angle of five minutes, the distance g would be 8 1/3 feet before the quantity would reach 0.0001-inch. Omitting $(g/\cos\ \theta) - g$ from Equation 1, w becomes twice the value of a, which is a close approximation for most purposes.

It should be remembered that the equations are valid only in seals where the stationary face is rigid. It might be pointed out, too, that if $f/\sin\theta$ is less than g the rotating face in the deflected position would be completely separated from the stationary one without any difference in axial position, and the equations would not be applicable. For $f/\sin\theta$ to be less than g, however, is an absurdity as far as machine design is concerned. The restriction is mentioned only as a purism. *Fig* 20 shows plots of a and w against θ for three different values of f.

Requires Axial Motion

These considerations indicate that if the stationary face is rigid, deflection should be held to the minimum and the rotating member should be free to move the required distance axially. It is true that the joint between the rotating member and the shaft might conceivably operate as a universal joint in the stationary face plane, but this places a cyclic load of rather high frequency on the material forming the joint, and introduces the limitation of fatigue failure. The amount of flexure is the quantity a, and the frequency is the angular velocity of the shaft. A quick calculation will show that with the best materials available for the purpose it is more practical to minimize deflection than to run the risk of fatigue failure in a few days of continuous operation.

Whip in a shaft, so far as mechanical seals are concerned, differs from deflection only in that it is exhibited every revolution of the shaft. In the case of deflection the geometric shaft axis coincides with the kinematic axis of rotation. In the case of whip the two do not coincide. When magnitude is appreciable it is usually caused by dynamic unbalance or a bent shaft. It is futile to attempt

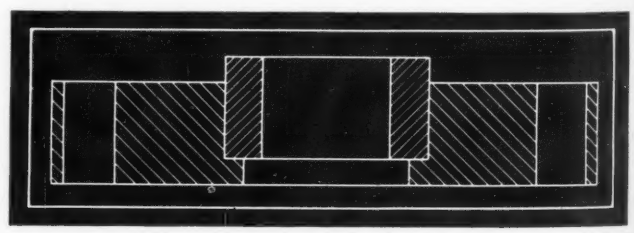


Fig. 24—Above—Preferred shoulder design because entire bushing length is used for resistance to shear

Fig. 25—Below—No provisions are made in this seal for pressing out bushing

to seal such a shaft except for the most simple conditions. Most all shafts have a small amount of whip due to changes in internal stresses in the metal subsequent to machining or hardening. These small amounts are not harmful because, translated into terms of w they do not interfere with most sealing requirements. It should be remembered, though, that a whip or deflection which does not cause a leak with lubricating oil, for example, might give serious trouble with propane or ammonia or similar fluids.

End play is not objectionable so long as the faces are kept in continuous intimate contact, because the very principle of a mechanical seal depends on axial freedom. The designer should consult with the seal manufacturer to make sure that the end play in his shaft is not greater in magnitude or frequency than that for which the seal is designed. In general, seals can cope with the amount of end play inherent in a new ball bearing.

Therefore, it is impractical to design elaborately for complete elimination of end play just for the sake of the seal. There are some types of seals, however, which are limited to only a few thousandths of end play, and with these it is safer to provide complete rigidity in the axial direction even though the contemplated end play is somewhat less than the seal will tolerate.

Stationary Member Alignment

The geometrical analysis used for shaft deflection or whip applies to stationary face misalignment also, except that θ becomes the angle between the axis of rotation and the stationary face. In seals whose stationary members are self-aligning this angle is zero as long as the rotating face rotates in a single plane. If the stationary member is not self-aligning the angle may still be made zero by means of the joint between the machine and stationary member. If the stationary member is fastened to the machine by a gasketed flange, for example, and the machine gasket seat is not normal to the axis of rotation, the bolts on one side may be pulled up tighter than on the opposite side, and the original misalignment is corrected.

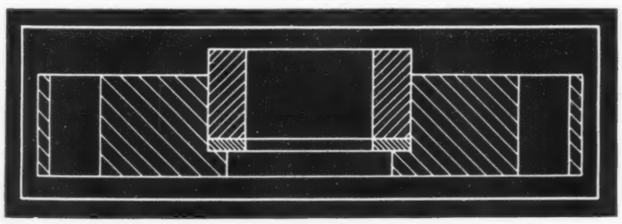
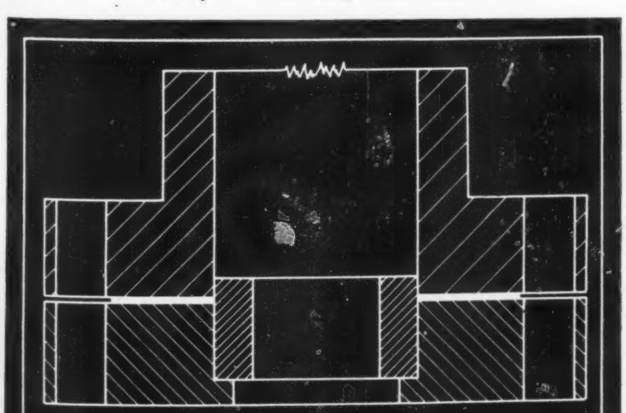


Fig. 26—Above—Washer puts bushing more in compression, less in shear, but still allows pressing out

Fig. 27—Below—Separate cylinder and flange are removed easily for refacing seal face



This is an unsatisfactory way of correcting the trouble, because the extent of misalignment is measured by the amount of leakage, and the best sealing takes place when there is no leakage from the very first revolution. Let it be supposed, though, that the machine gasket seat is normal to the axis of rotation. There is still no assurance that the face of a rigid, gasketed stationary member will be normal to the shaft because the gasket may be compressed unevenly. In other words, the gasket between the machine and a rigid stationary member may either correct misalignment or cause it. The first is objectionable because of its uncertainty, and the second is objectionable per se.

The apparent solution would be to avoid the use of rigid stationary members which do not preserve correct alignment through metal to metal contact. It has been found that, because of their ability to take high thrust loads, rigid stationary members are well adapted to sealing high pressures and are not limited in choice of materials for heat and corrosion resistance; but the standard joint, between easily replaceable rigid stationary members and the machine, is a gasket. The seal manufacturer favors the gasket because the seal is nearly always applied to equipment not "built around the seal", and the ability of the gasket to correct misalignment is more important than its likelihood of causing it. Actually, therefore, one cannot always solve the problem simply by avoiding that type of seal. It would seem that the next move is the seal manufacturers', although the use of O-rings instead of gaskets in the autoclave bearing assembly shown in *Fig.* 18 suggests a solution in cases where the fluid will not soften the O-ring material by heat or chemical attack.

Wearing-in Time Should Be Minimum

Mention was made earlier that the best sealing takes place when there is no leakage from the very first revolution The reason for this is that leakage might carry abrasive particles between the faces and cause rapid wear or scoring. Seals which must "wear in", that is, seals with rigid stationary members, leak until they have worn to a seal. Anything which can be done to reduce this wearing-in time will be reflected in longer seal life from the standpoint of wear and also from the standpoint of sudden failure due to scoring. Self-aligning stationary members present no problem in this respect, but where rigid ones must be used not only should the machine gasket seat be normal to the axis of rotation but care should be taken to avoid warping of the stationary member when it is pulled up.

Ordinary stuffing boxes are provided with two diametrically opposite studs or bolts for taking up the gland. When these studs or bolts are used for fastening a rigid stationary member to the machine and the pressure on the gasket between them must be high, there is great danger of warping the stationary face. Contact with the rotating face is then centered near a line connecting the two studs

or bolts. Liquid enters between the faces elsewhere, perhaps carrying abrasive particles which are then wedged between the faces at the points of contact. *Fig.* 21 shows such a stationary member, without its gasket. The score marks are clearly visible where contact was made, and the other portions of the surface are untouched. This seal would have worn out before it wore in. The addition of two more bolts resulted in acceptable performance, although the basic problem of initial alignment remained.

Accessibility for Installation and Maintenance

A designer laying out a machine in which seals are to be used can easily "paint himself into a corner" if he doesn't bear in mind the fact that seals, unlike packing, are endless rings without a split. They are mounted only by moving them over a free end of the shaft. Split seals have been used experimentally in special instances but there is little to recommend the practice. If it appears that there is no way to seal a shaft except by splitting a seal around it one would be wise to start with a new seal design rather than to try splitting a seal which was designed originally without a split.

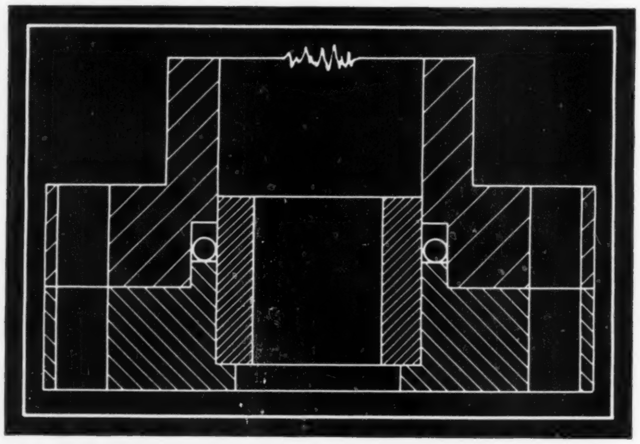
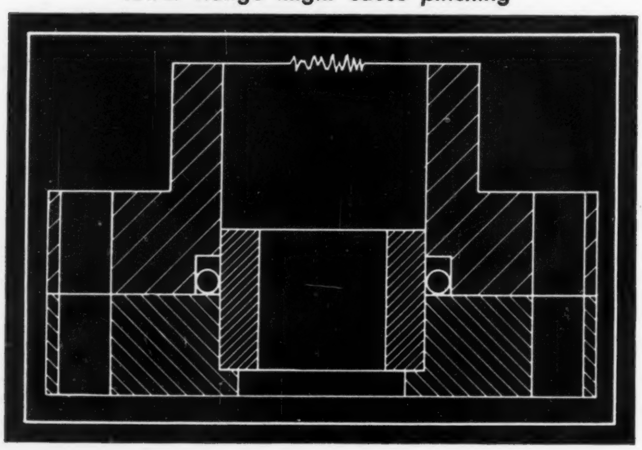


Fig. 28—Above—O-ring permits metal to metal contact for better alignment

Fig. 29—Below—Installation of O-ring without step in lower flange might cause pinching



One should remember, too, that between the point where the rotating member fits on the shaft and the shaft end there must be no steps of larger diameter than the seal bore. And if the rotating member is a type which seals on the shaft, the shaft surface at that point should be smooth and preferably polished. It is often possible, too, to utilize existing steps or shoulders on the shaft for positioning the rotating member. Any device which makes careful measurement unnecessary for installation is desirable.

Design of Shoulder Is Important

One might take, for example, the problem of incorporating a rigid stationary member in a machine. To do this the bearing material is usually pressed into a retainer ring which is then bolted to the stuffing box or housing. One design for such a pressed fit is shown in *Fig.* 22. Even though the shoulder is thick enough to resist shearing after the bearing section is pressed home against it with the pressure necessary to make the fit, the average machinist will be uneasy when he feels the piece going home. He hesitates between giving it one final push to make sure it's squared up, running the risk of shearing or crumbling the shoulder, or leaving it the way it seemed to go home, taking a chance that the shoulder is touching evenly all around. This is particularly so when the bushing is at the bottom of a cavity, *Fig.* 23, and visual inspection of the shoulder contact is impossible. And when the material is carbon or graphite, or the shoulder thickness has been reduced by refacing cuts during repairs, the hazard is by no means imaginary.

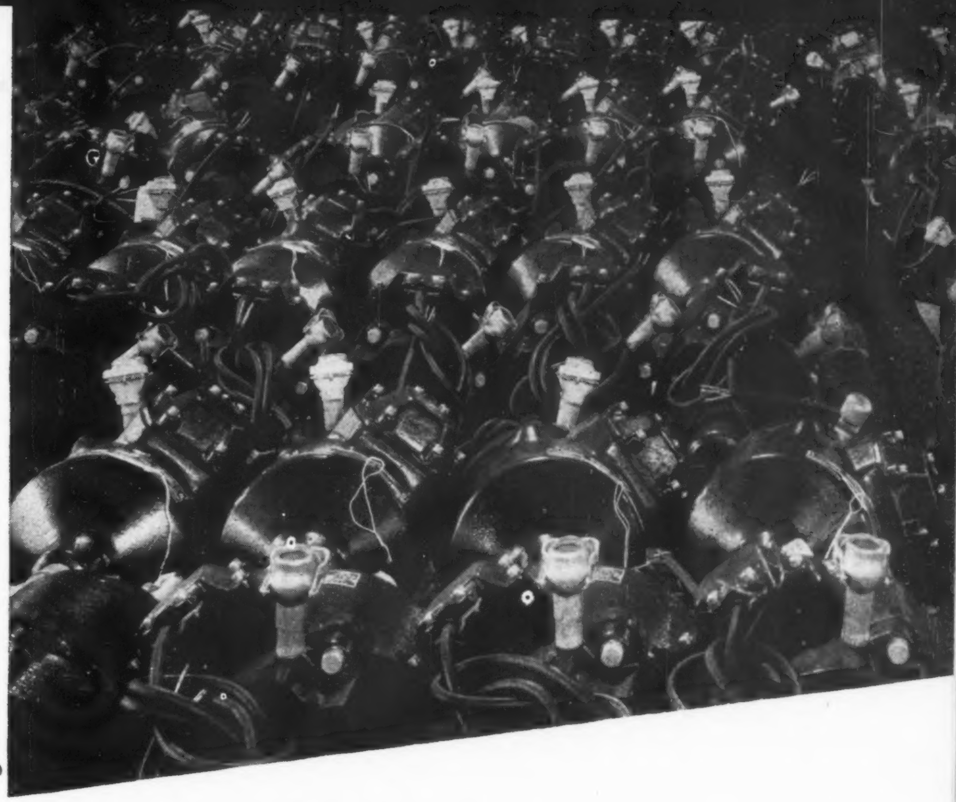
A better design is shown in *Fig.* 24. The whole length of the bushing is now available for resistance to shear. The machinist feels free to give it the final push. One should remember, though, not to close the bottom of the counterbore down to the shaft, as in *Fig.* 25. The only advantage is that the material is now in compression rather than shear, but the machinist has no way of pressing the piece out. Ordinarily he would cut it out, but there is always the possibility of an emergency when the piece must be used again. If it is imperative that the bushing be in compression with very little shear a washer may be used, as in *Fig.* 26.

Provides for Refacing

When the bearing surface is at the bottom of a cavity, as in *Fig.* 23, the best refacing job cannot always be done on it. The piece is ordinarily chucked in a lathe, and a light finishing cut is taken across the face after all wear or score marks have been cut down. A single point tool leaves the best finish when it is most free from vibration. Reaching down into the cavity requires tool overhang, with consequent increase in chatter. If consideration is to be given to refacing an arrangement such as is shown in *Fig.* 27 is advisable. The addition of a flange to the cylindrical section permits its removal, making proper refacing of the stationary member much easier. Many seal users make it a practice to lap the stationary surface, and in such instaces the advantage of the design in *Fig.* 27 over that in *Fig.* 23 is immediately apparent.

If the upper end of the cylindrical piece in *Fig.* 27 is to be used to support a bearing or some other element which requires precise alignment, the gasket introduces the pos-

(*Concluded on Page* 182)



NEW Fractional Horsepower Motor Standards

Recent NEMA standards give assurance to designers that motors received from different manufacturers will have similar characteristics

TO THE engineer concerned with the application of fractional-horsepower motors to machines, the need for standardization is so evident as to require no comment. That the standards in use to date have been inadequate is a well-known fact. The new National Electrical Manufacturers Association standards are a recognition of the deficiency of the old standards and are a long stride toward simplification of design. With the increase in use of small motors in such new applications as typewriters and remote drive, the new NEMA standards should be of great help to manufacturers as well as designers and development engineers.

Since breakdown torque has been defined as the maximum torque the motor will develop without abrupt change of speed, a motor should be selected to offer a breakdown torque sufficient to take care of the maximum peak load requirements of the application concerned. In order to cover the various horsepower ratings, the new standards have divided motors into groups with respect to speed and power. This standard is shown in Table I.

General-purpose motors have a low temperature rise rat-

Table I

Breakdown Torques for Single-Phase Induction Motors

	Revolutions Per Minute					
Synch. RPM	3600	3000	1800	1500	1200	900
Approx. F.L. RPM	3450	2850	1725	1424	1140	850
Rating (bhp)	Breakdown Torque (Oz-Ft)					
1/20	2.0- 3.7	2.4- 4.4	4.0- 7.1	4.8- 8.5	6.0-10.4	8.0-13.5
1/12	3.7- 6.0	4.4- 7.2	7.1-11.5	8.5-13.8	10.4-16.5	13.5-21.5
1/8	6.0- 8.7	7.2-10.5	11.5-16.5	13.8-19.8	16.5-24.1	21.5-31.5
1/6	8.7-11.5	10.5-13.8	16.5-21.5	19.8-25.8	24.1-31.5	31.5-40.5
1/4	11.5-16.5	13.8-19.8	21.5-31.5	25.8-37.8	31.5-44.0	40.5-58.0
1/3	16.5-21.5	19.8-25.8	31.5-40.5	37.8-48.5	44.0-58.0	58.0-77.0
1/2	21.5-31.5	25.8-37.8	40 5-58.0	48.5-69.5	58.0-82.5	....
3/4	31.5-44.0	37.8-53.0	58.0-82.5	69.5-99.0	....	....
1	44.0-58.0	53.0-69.5	....	....	....	....

Breakdown torque range includes the higher figure, down to, but not including the lower figure.

Table II

Service Factor for General Purpose 40 C Induction Motors

Rated HP	Factor
1/20	1.4
1/12	[illegible]
1/8	1.4
1/6	[illegible]
1/4	1.35
1/3	1.35
1/2	1.25
3/4	1.25
1 at 3600 RPM only	[illegible]

Table III

Locked Rotor Current Single-Phase Motors

Rated Hp	Amperes at: 115 Volts	Amperes at: 230 Volts
1/6 and smaller	20	10
1/4	23	11.5
1/3	[illegible]	
1/2	45	22.5
3/4	61	30.5

ing compared to motors designed for specialized duty. The general-purpose motors are rated at 40 C rise continuous duty, compared to the specific-purpose motors with ratings of 50 C rise if open and 55 C rise if totally enclosed. It has been recognized, however, that fractional-horsepower motors may be operated safely at loads exceeding nameplate ratings. Owing to physical size and breakdown torques, the small motors are capable of operating at overload without exceeding the specified 10-degree temperature rise margin of safety. This overload capacity when taken as a factor of rating is known as service factor. As adopted by the NEMA, service factor ratings, shown in Table II, define the loading that will give a total temperature rise of 50 C to general-purpose fractional-horsepower motors having 40 C rise nameplate rating.

Maximum locked-rotor current specification is not new, having been adopted some years ago. This standard provides that the locked-rotor current of 60-cycle single-phase motors in the 900-3600 rpm range, excepting split-phase motors now used on washing and ironing machines, shall not exceed values given in Table III.

The new standardization also extends classification to stamping a code letter as designated in Table IV on the nameplate of motors rated to 1/20-horsepower. This provides a marking from which can be determined the actual starting current and conformance with the standard.

Table IV

Locked Rotor Starting Current

Code Letter	KVA per Horsepower*
A	0- 3.15
B	3.15- 3.55
C	3.55- 4.0
D	4.0- 4.5
E	4.5- 5.0
F	5.0- 5.6
G	5.6- 6.3
H	6.3- 7.1
J	7.1- 8.0
K	8.0- 9.0
L	9.0-10.0
M	10.0-11.2
N	11.2-12.5
P	12.5-14.0
R	14.0- and up

* Locked kva per horsepower includes the lower figure, up to but not including the higher figure.

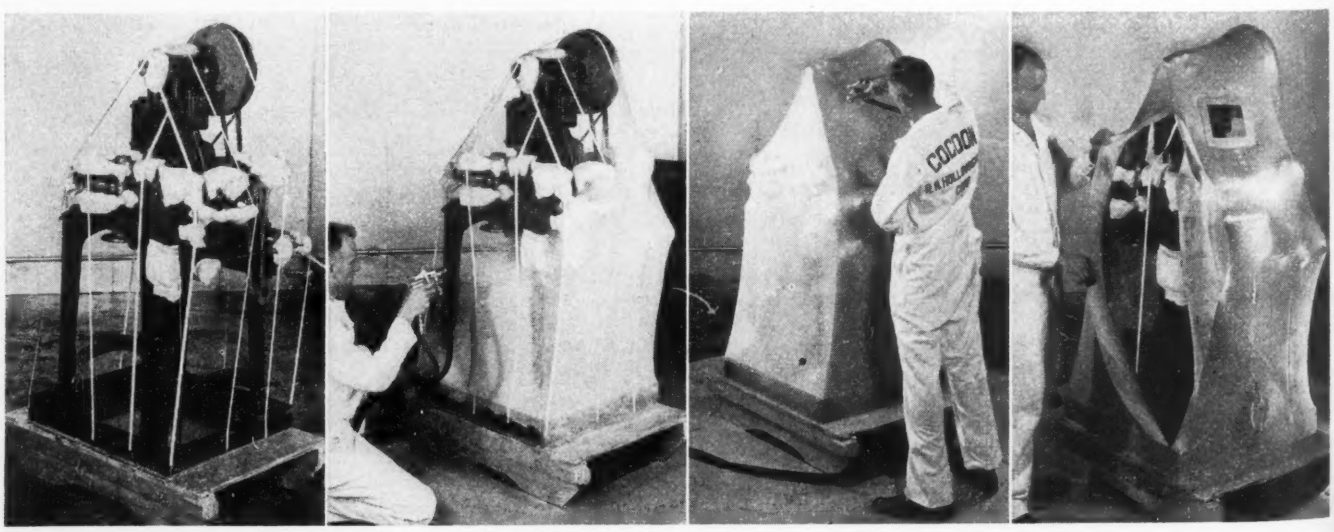
Cocoon for Machine Packaging

INTERESTING POSSIBILITIES for packaging or scup coating lie in an improved plastic developed by the Hollingshead Corp. The coating consists of a vinyl compound which is sprayed over an entire machine or part, protecting it from handling and the elements.

The method of application is shown in the series of illustrations below. Sharp projections on the machine are protected with fabric pads, and a latticework of scotch tape is applied to serve as a framework for the plastic. Next, a mixture of vinyl acetate and modified copolymer vinyl resins with plasticizers and webbing agent is sprayed over the latticework. This base coat is of only sufficient strength to carry subsequent coatings.

Over the base coat are sprayed three layers of plastic material identical with that first applied, with the difference only that the webbing agent has not been added. In each case the coat is of sufficient thickness as to completely hide the preceding layer.

Total thickness of the plastic coating is approximately 0.040-in. It has a tensile strength of about 2000 psi, elongation at room temperature of 200 per cent, and does not become soft or tacky at 180 F.



MACHINE *Editorial* DESIGN

Engineering Is Best Insurance

ONE of our top Air Force generals has predicted that if there should be another war it would last only 40 minutes. Two developments of World War II—the atomic bomb and the guided missile—conceivably could, in the hands of a determined aggressor, be developed to the point where their use in a surprise attack would virtually paralyze the nerve centers of a country before any adequate defense could be mobilized.

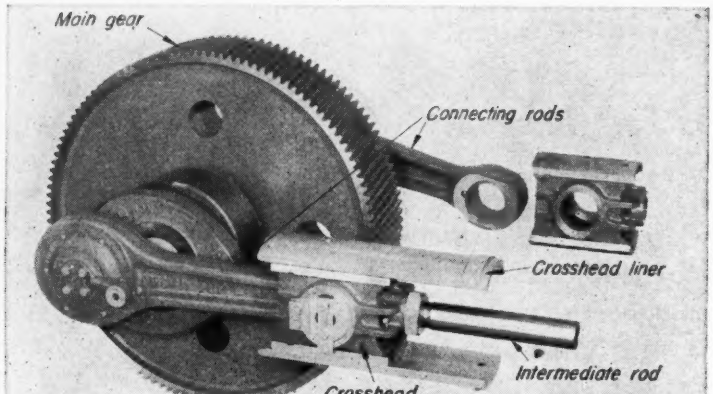
Thus a fleet of bombers having range and capacity comparable to those of the B-36, loaded with atomic bombs which could be guided to their targets by remote control, seem to offer an obvious strategy. Less fully appreciated, perhaps, are the potentialities of a stream of atomic-fuelled rocket projectiles with atomic warheads, which would soar hundreds of miles above the earth and rain down on cities thousands of miles away.

In their present stage of development, rockets and guided missiles might strike terror into whole populations but do not offer the immediate prospect of a 40-minute "pushbutton" war. Radio controls can be too easily jammed, while unguided rockets so far have not been notably accurate. This is not to say, however, that intensive research and development on weapons of this type will not eliminate or greatly reduce such deficiencies.

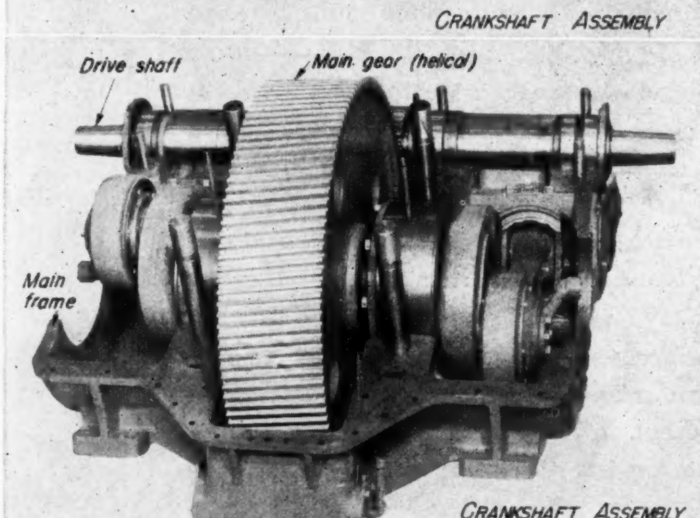
Although every effort should continue to be made to set up a workable international organization for the prevention of war, including the proper control of the manufacture of fissionable materials, it is much too soon to place complete faith in a system which has yet to be proved infallible. Even though our intentions are strictly peaceful, we should nevertheless keep our lead in the development of new weapons, as well as all possible protective countermeasures.

To insure that military research and development, as well as parallel activities in industrial laboratories and engineering departments, and in universities, shall go on unabated, it is to be earnestly hoped that Congress will have the wisdom to lend wholehearted support. This means not only adequate financing but also legislation that will enable student engineers and scientists to complete their education without interruption from any possible source.

DESIGNS OF THE



CRANKSHAFT ASSEMBLY

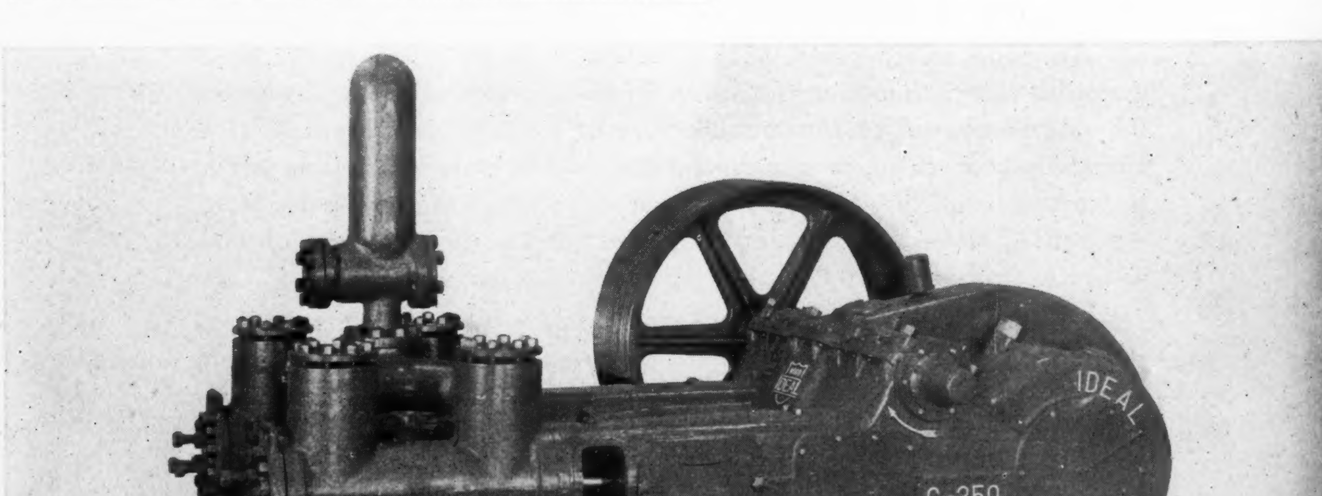


CRANKSHAFT ASSEMBLY IN MAIN FRAME

Power Slush Pump

FRAME and frame cover of this pump are alloy cast iron, with cover serving as main bearing and pinion-shaft bearing caps. Thus, removal of cover permits both pinion shaft and crankshaft to be lifted out or set in with all component parts assembled. To hold deflection of the pinion shaft to a minimum, it is mounted in four roller bearings. The 9-inch diameter crankshaft is of overhung design and is supported by two matched single-row taper roller bearings on each side between gear and crank. Solid cast-steel cranks are shrunk and keyed onto the crankshaft, no bolts being used, and the crankpin is cast integrally with the crank.

Helical main gear and driving pinion are employed, the speed reduction being 5.64 to 1. The pinion is a heat treated steel forging, shrunk and keyed onto its shaft. The main gear is a patented welded assembly, heat treated and pressed and keyed onto the crankshaft. Connecting rods are solid I-section steel castings with no splits or bolts. Bearing for each crankpin comprises two matched single-row tapered-roller bearings, and the crosshead bearing is a bronze bushing in which operates a hardened and ground steel crosshead pin that is keyed to the crosshead to prevent rotation. Crossheads are fitted with replaceable cast iron slippers which can be adjusted by means of shims. Cast-iron crosshead guides, bolted in the frame, also are

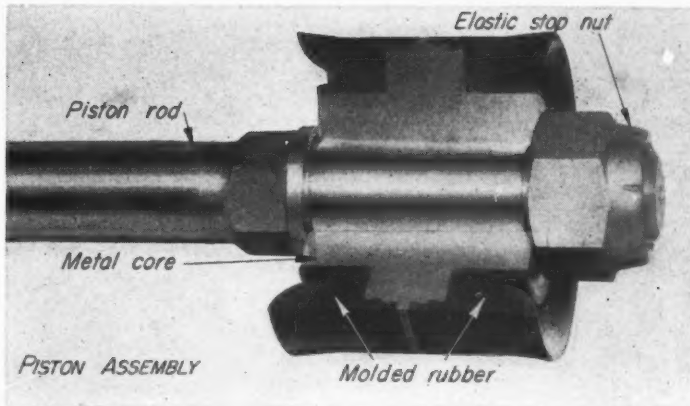


ONTH

replaceable. The intermediate rods screw into the crossheads and are held tightly by lock nuts.

No mechanical lubricator is used on this pump. Lubrication of the main gear, pinion and crossheads is by the splash system, oil being picked up from the sump by the main gear, collected under the pinion and cascaded over the crossheads. The sump is provided with an oil-level indicator and a cleanout plate fitted with a magnetic drain plug. All roller bearings are greased packed and sealed to prevent entrance of crankcase oil.

Patented one-piece pistons employed are of tough, resilient rubber molded on a solid metal core, and the seal between piston and rod is effected by an accurate taper fit. Piston rod is alloy-steel forging, heat treated and ground. Fluid end of the pump is one-piece alloy-iron casting. Suction passages are direct, smoothly curved and of large area for minimum resistance to fluid flow. Each fluid-control



PISTON ASSEMBLY

valve is in a separate cavity, and there is a positive metal-to-metal seal between each valve seat and its bore in the fluid-end casting.

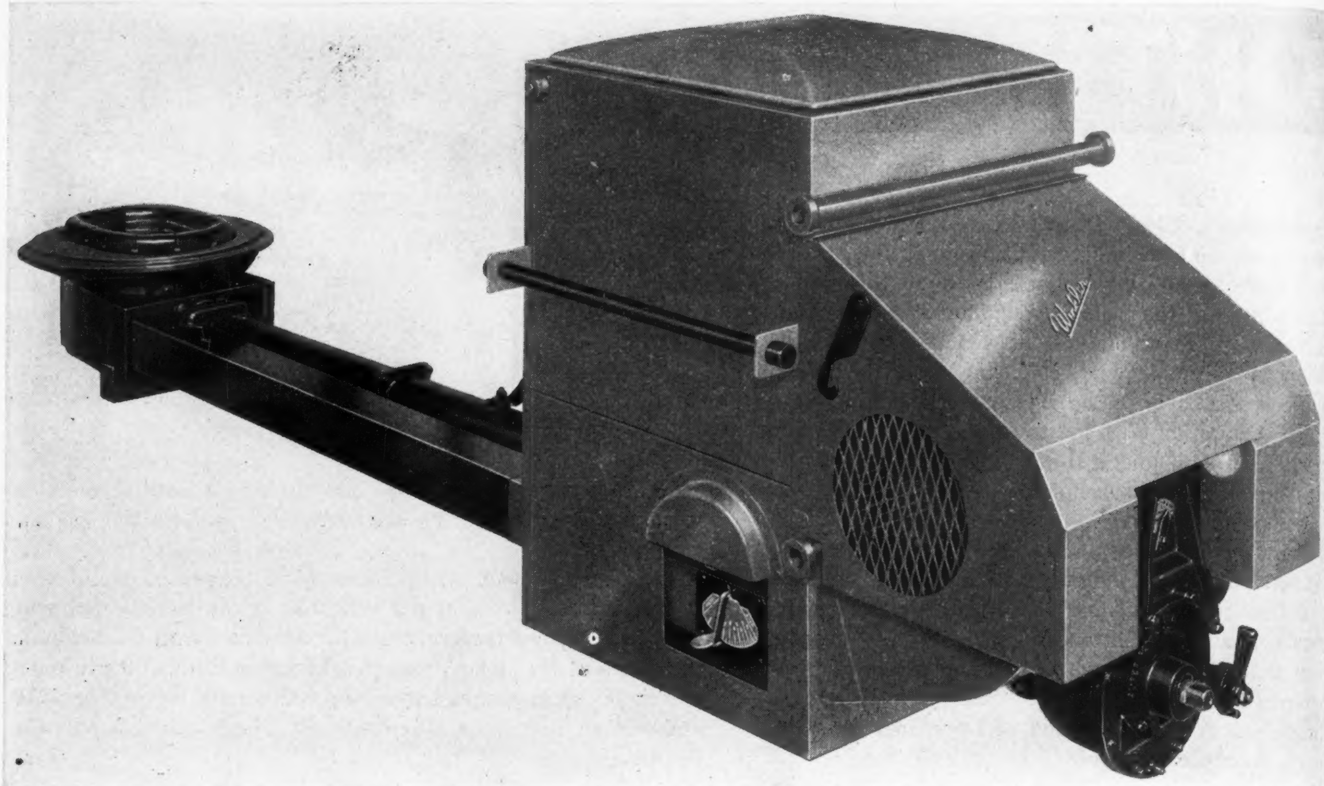
Based on mechanical efficiency of 85 per cent and volumetric efficiency of 100 per cent, this pump has the following specifications and performance characteristics with fluid-cylinder liner bore of 7¼ inches: Max. discharge pressure, 700 psi; pump speed, 65 spm; nominal input hp, 320; rated output hp, 270; and volume, 660 gpm. Manufacturer: The National Supply Co., Pittsburgh.

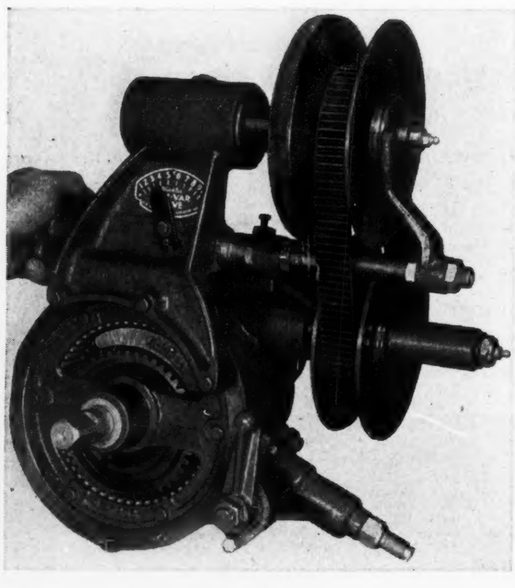
Automatic Stoker Transmission

IMPORTANT requirements of stoker transmissions are that they effect a large speed reduction between drive motor and feed screw and, at the same time, act as efficient force multipliers. The transmission shown is cross section meets these requirements in the following manner:

Drive from the motor is to the drive pulley which, being keyed to the worm shaft, rotates the worm at high speed. The worm engages a worm gear which is fastened to an eccentric which turns freely on needle bearings over the main drive shaft. Mounted over the eccentric on needle bearings is a planet gear which meshes with two ring gears, mounted side by side. One of these ring gears has 54 teeth (left), the other 55. Obviously when the planet gear revolves within the ring gears, these gears must change their respective positions by one tooth (1/55-revolution) each time the planet gear makes a complete

SECTIONS THROUGH STOKER TRANSMISSION





revolution. Both ring gears have a tendency to turn because of the action of the planet gear. However, the 54-tooth ring gear is locked against rotation by a detent pin which is pressed into one of three pockets in the periphery of the gear by a compression spring. The other ring gear (55-tooth) is splined to the main drive shaft, which it drives at the required slow speed

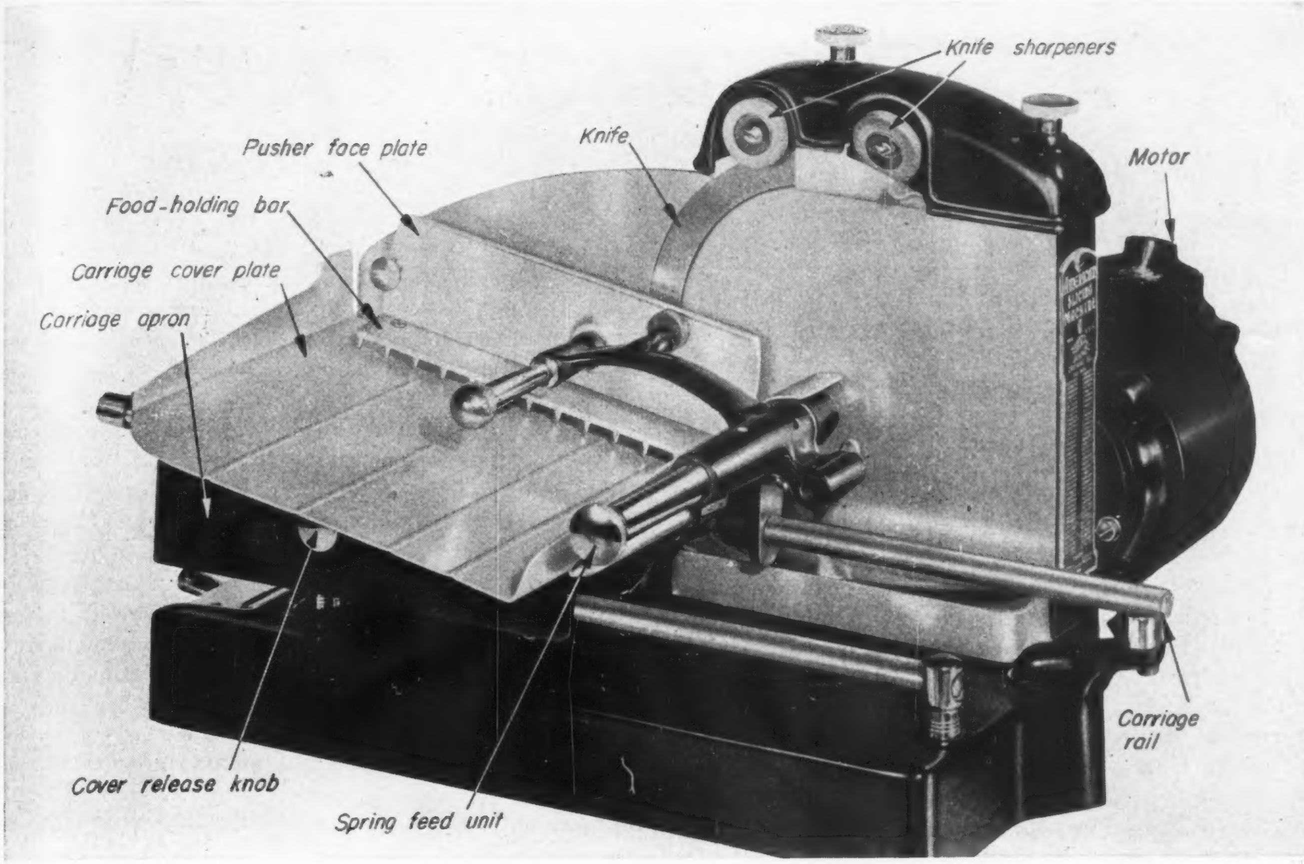
This stoker employs no shear pins. When the load on the feed-screw increases to a predetermined value because of an obstruction in the feed-screw tube, the increased turning effort transmitted to the 54-tooth ring gear will overcome the pressure of the detent spring. This, in turn, will allow the detent pin to slide out of its locking pocket, thereby releasing the driving mechanism from the feed screw. After the obstruction has been removed, the detent automatically re-engages, re-establishing the normal drive of the feed screw.

Means is provided for releasing the feed screw from the driving mechanism manually. A lever is attached to a short shaft on the opposite end of which is a small eccentric that engages with a special slot in the detent pin. Moving the lever to the "off" position forces the detent pin out and holds it in that position until the lever is returned to the "on" position. Entire thrust of the feed screw is taken by an over-sized combination radial and thrust ball bearing; no thrust being taken by the transmission cover plate.

On one model of this stoker (see photograph of transmission) the drive from motor to worm shaft is by belt to a pair of variable sheaves which are adjustable by the screw-positioned forks shown. The remainder of the transmission, however, is the same as that described above. Manufacturer: U. S. Machine Corp., Lebanon, Ind.

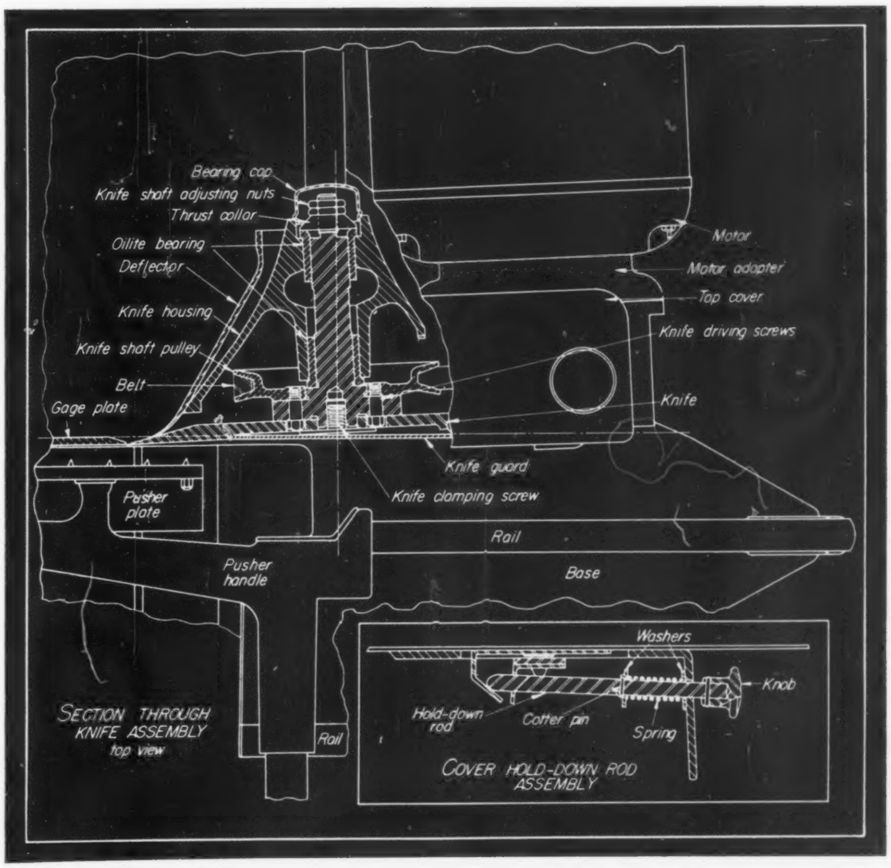
Slicing Machine

ROTATING circular knife of this slicer is set at an angle, permitting only the honed edge to contact food. As the cross section drawing through knife assembly shows, drive is by V-belt and knife shaft runs in self-lubricating Oilite bearings. Wheel-type sharpening stones are built-in, being moved into sharpening position by conveniently located lever. The carriage assembly rides on steel rails which are bolted to the cast-iron main chassis. Carriage cover is removable, being held in place by a spring-loaded rod as shown in the hold-down rod assembly drawing.



Pusher face plate employed is anodized aluminum and employs a double-tooth bar for holding food firmly in place while slicing. Feed into the knife is effected by a telescoping-spring unit designed to provide substantially uniform pressure throughout entire slicing operation. Power for driving knife is supplied by a ¼-hp motor mounted on the main chassis and operated by a simple on-off switch located on the motor housing.

On the receiving side of the machine (not shown) there is an anodized-aluminum base shield which fastens to the main chassis and is held firmly by spring bolts. Also there is an anodized-aluminum receiving tray and motor shield which fasten by slide clips to extended steel bars bolted to the main chassis. An anodized-aluminum deflector is mounted on the knife shaft housing with a swivel bolt. Manufacturer: American Slicing Machine company, Chicago.



DESIGNS OF THE MONTH

ASSETS to a BOOKCASE

Modern Turbines

By Newman Keller, Lyons and Wales all with the General Electric Co., published by John Wiley and Sons, Inc., New York; 175 pages, 5⅝ by 8⅝ inches, clothbound; available through Machine Design, *$2.50 postpaid.*

This book has been written for the engineer concerned with the application of turbines rather than with their design. To this end, the authors after briefly discussing the basic principles of turbines concern themselves with the external characteristics of the units. Covering steam turbines only, the authors discuss power plants, selection of turbines, performance characteristics, and estimation of steam performance.

The book may be divided into three basic sections: Part one deals with turbines, describing them and comparing the types. Part two concerns characteristics and performance with a discussion of turbine accessories and the analysis of performance curves. Part three discusses governors and electrical aspects of turbine usage.

□ □ □

Making Patent Drawings

By Henry Radzinsky, Patent Attorney, published by the Macmillan Co., New York; 96 pages, 7½ by 10¼ inches, clothbound; available through Machine Design, *$3.00 postpaid.*

While the typical engineer-inventor will rarely be called upon to make his own patent drawing, it will still fall within his province to criticize and correct drawings made for him by patent draftsmen. In the light of this fact it behooves the inventor to acquaint himself with the demands made by the patent office and the limitations made upon the art of illustrating.

In this book, the author discusses in detail the requirements of the drawing. He points out what should be shown and what should not be shown, mentions the use of detail and of schematic, the use of views etc.

□ □ □

Experimental Stress Analysis

By the Society For Experimental Stress Analysis, published by the Addison-Wesley Press, Inc., Cambridge 42, Mass.; 166 pages, 8¾ by 11¼ inches, clothbound; available through Machine Design, *$5.00 postpaid.*

This is the sixth in a series of compilations of stress analysis papers prepared by members of the Society for Experimental Stress Analysis. While of basic interest to students of theoretical mechanics, this book will be found of profound value to any engineer engaged in design involving stress computations, for needless to say the basic research of yesterday provides today's design data.

Of particular interest to the designer are sections concerning: Stress in spur gear teeth, mechanical failures of steel plant equipment, allowable working stresses as determined by physical properties, and fatigue failure of manufactured parts.

The paper discussing allowable working stresses offers data of unique value in the form of 28 tables and curves. Given are tensile-hardness relationships, maximum-mean working-stress curves for materials of various hardnesses, and other charts of equal value. As with other books in this series, this book carries the evident mark of careful editing and shows thought in the broad selection of material.

Flash Welded Joints

Translated from the German by men of the National Defense Research Council at Battelle Institute, the recently published booklet "Production Technique and Quality of Flash Welded Joints" by Hans Kilger is said to be the most complete publication on flash welding that has appeared in literature to date. Although limited in this work to manually operated machines and to the simpler commercial carbon steels, the author has considered and partially answered many of the problems confronting industry today. Data on effect of temperatures and upset pressures on the quality of flash-welded joints, and the method for obtaining these data, are particularly instructive and should provide a basis for further research along these lines. The booklet is available from the American Welding Society, 33 W. 39th St., New York 18; price $1.00.

□ □ □

Welding Aluminum

Manual on welding procedure and factors for aluminum published under the title "Welding Aluminum" has recently been announced by the Reynolds Metals Co. The manual discusses the more widely used aluminum alloys from the standpoint of weldability factors and analyzes in some detail recommended gas and arc welding practice. It covers such topics as edge preparation cleaning and preheating, fluxes, seam and flash welding, brazing and soldering. The paper-bound book is available from Reynolds Metals Co., Department 47, 2500 S. Third St., Louisville 1, Ky., for $1.00.

Design of Fluid-Film Bearings for Optimum Loading

By H. A. S. Howarth
Bearing Engineer
Philadelphia

Part II—For Width-Diameter Ratios Other than Unity

PERFORMANCE formulas, tables and a chart for 120-degree centrally loaded clearance bearings with width-diameter ratio of unity were presented in last month's data sheet. The present data sheet extends the information to cover ratios other than unity. In addition, as a guide to the proportioning of bearing diameters and widths, current design practice as it concerns pressures and velocities is summarized on a *PV* chart.

Equations 1 to 4 on *Fig.* 1 of last month's data sheet are applicable, as already stated, only to bearings having a length-width ratio of unity. The formulas for h_0 and c, however, apply regardless of ratio while the formula for h is general to bearings with running clearance and the one for V in terms of N and d is a standard fundamental relationship.

For 120-degree centrally loaded clearance bearings of any length-width ratio Equations 1 to 4 may be rewritten

$$\frac{Pa}{V\mu}\left(\frac{\eta}{a}\right)^2 = 0.228\ L_1 \qquad (1a)$$

$$\frac{H}{PV}\left(\frac{a}{\eta}\right)\frac{10^6}{a} = 9.36\frac{F_1}{L_1} \qquad (2a)$$

$$\frac{\lambda}{\left(\frac{\eta}{a}\right)} = 1.542\frac{F_1}{L_1} \qquad (3a)$$

$$\frac{\lambda}{\sqrt{\frac{\mu N}{P}}}\ 0.533\frac{F_1}{\sqrt{L_1}} \qquad (4a)$$

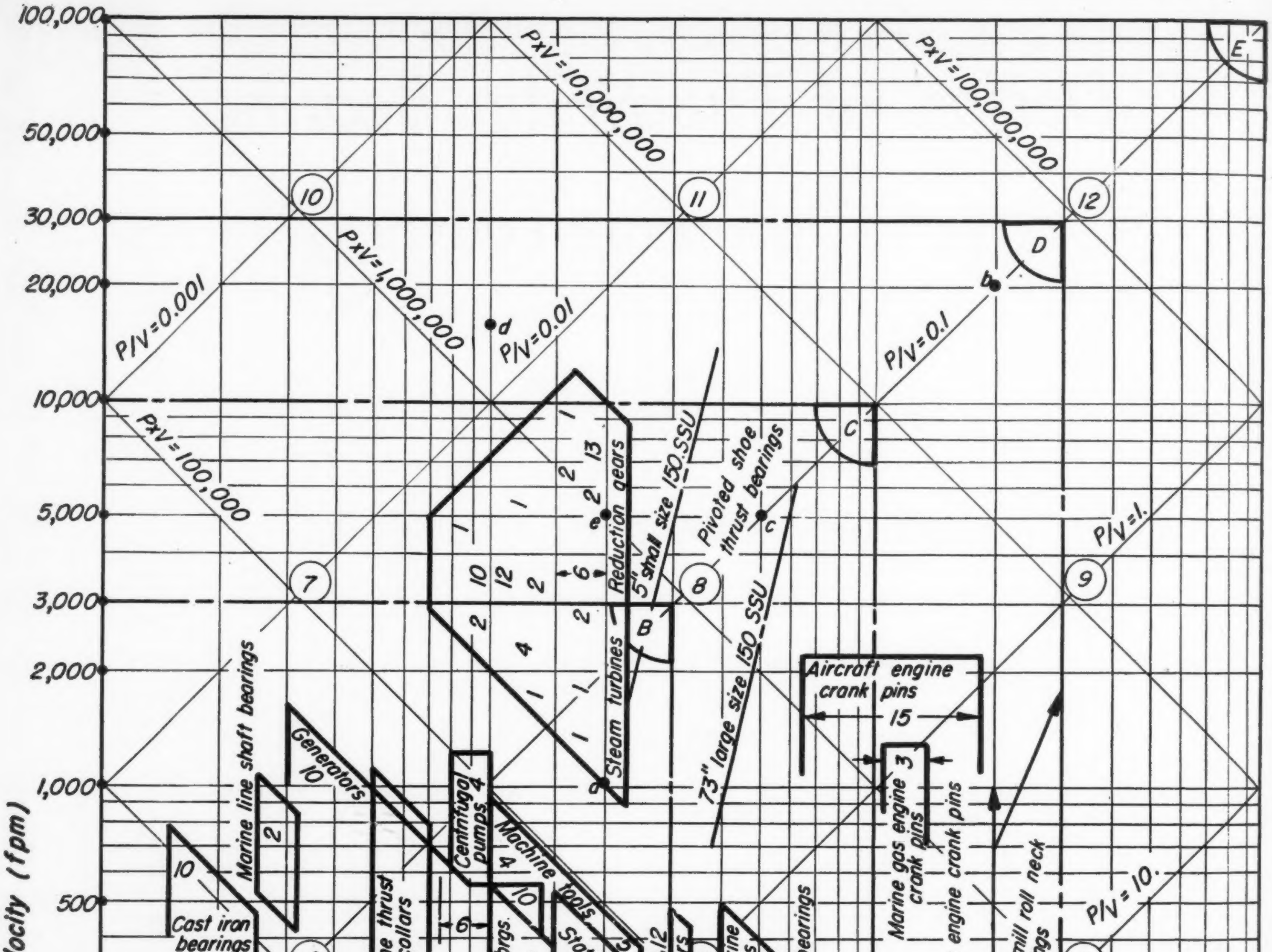
where F_1 and L_1 are side leakage correction factors for friction and load, respectively. Numerical values for F_1 and L_1 appear in TABLE III, which also includes the resultant obtained by evaluating the right-hand side of each of the four equations. Inasmuch as $l/d = 1.047$ for the 120-degree bearing, for practical purposes it may be assumed that $l = d$ and therefore $b/d = b/l$. The last two columns of TABLE III give the approximate width and area of the bearings on this assumption.

Next question that arises is "How shall the least thickness table, TABLE I, be used with the l/b proportions covered by TABLE III?" It is believed reasonable to use the same least thickness for all bearings having the same projected area and the same surface speed. Thus, a 9-inch diameter journal whose width of bearing is 13½ inches has l/b ratio 0.667. Projected area of the journal will be 121.5 square inches, the square root of which is 11.022 inches, hence the same least thickness may be used for the 9 by 13½ bearing as for a bearing measuring 11 by 11 when the surface speeds are the same. This method will apply whether the length-width ratio is less or more than unity.

EXAMPLE: To show how TABLE III may be used for solving bearing problems where the length-width ratios are other than unity, assume $P = 200$, $V = 5000$, $d = 9$ inches and $b = 16$ inches.

SOLUTION: Speed of rotation is $N = V/0.262d = 2120$ rpm. Length-width ratio will be approximately equal to the diameter-width ratio, which is 0.5625. This value lies between 0.500 and 0.667 in the l/b column of TABLE III, and the tabular values for the formulas can be found by interpolation.

Projected area is $9 \times 16 = 144$ square inches, the square root of which is 12, hence the least film thickness, from TABLE I, is $h_0 = 0.00288$ for $V = 5000$ and $d = 12$. From this it follows that $\eta = 0.00542$ and $\eta/a = 0.0012$. For Equations 3*a* and 4*a* the tabular values will be, respectively, 2.24 and 0.628, from TABLE III. Then $\lambda = (\eta/a) \times 2.24 = 0.0012 \times 2.24 = 0.00269$ and $(\mu N/P)^{1/2} = \lambda/0.628 = 0.00269/0.628 = 0.00428$, from which $\mu N/P = 18.3 \times 10^{-6}$. Inasmuch as $P/N = 200/2120 = 0.0943$, $\mu = 18.3 \times 10^{-6} \times 0.0943 = 1.726 \times 10^{-6}$ reyns.



Velocity (fpm)
Pressure (psi)
PxV=1,000
PxV=10,000
PxV=100,000
PxV=1,000,000
PxV=10,000,000
PxV=100,000,000
P/V=0.001
P/V=0.01
P/V=0.1
P/V=1.
P/V=10.
P/V=100
Cast iron bearings
Marine line shaft bearings
Generators 10
Marine thrust collars
Centrifugal pumps 4
Machine tools
Ordinary brgs. bronze or Babbitt 10,12
Stationary engines main bearings
Rwa. cars
Stationary & marine engine crank pins
Marine engine main bearings
Gas engine main bearings
Steam turbines
Reduction gears
5" small size 150 SSU
73" large size 150 SSU
Pivoted shoe thrust bearings
Aircraft engine crank pins
Marine gas engine crank pins
Gas engine crank pins
Steel mill roll neck bearings
Stationary overhung crank pins
Loco. crank pins
Slow running machine tools 5

Fig. 4—Pressure-velocity chart for bearing performance, divided into twelve fields. Areas are indicated for various types of bearings. Areas from A to E are indicated as zones of conservatism. Uncircled numerals refer to references listed on Page 154

ENGINEERING DATA SHEET

and $Z = \mu \times 6{,}900{,}000 = 11.9$ centipoises.

Power loss, from Equation 2*a* for which the tabular value is 13.6, will be $Hb = 11.75$ hp. Load carried by the bearing at 200 psi on 144 sq in. projected area is 28,800 lb.

PRESSURE-VELOCITY CHART: As an index of probable performance of a bearing, the product of pressure and velocity, *PV*, often is employed in preliminary design. A chart on which are recorded such data from bearings used on a variety of machines is shown in *Fig.* 4. The chart is of the log-log type with twelve fields, each of which is identified by an encircled number, 1 to 12. Pressure *P* is expressed in pounds per square inch of projected area of the journal, i.e., journal diameter *d* times bearing width *b* measured axially. When points on the chart refer to thrust bearings, *V* would be the mean velocity in feet per minute and *P* would be the average pressure per square inch upon the net supporting area. Because of the difference of meaning of *P* for the two kinds of bearings it would, perhaps, be better to put thrust bearing data on a chart by itself. It is even precarious to put all journal bearing data on one *PV* chart, because the actual average pressure throughout a journal bearing film will vary with the angular length of the film. If the working length of such a film were always equal to, say, two radians it would be possible to compare data more effectively.

A *PV* chart whether of the log-log or some other type possesses a rather limited value, in spite of its appearance of importance, because a point on it shows merely a combination of pressure and velocity. The point does not give the journal diameter, the clearance ratio, or the viscosity of the lubricant. Nor does it give the bearing type, its proportions, or the side leakage correction factors. Therefore, in spite of the wide range of information it contains of the bearing design habits of different industries, it is still deficient as to vital design data. Enlargement of the chart and separation of data into like classes for a given chart would make possible the tabulation at each point of data vital to each application.

Regions outlined and described on *Fig.* 4 were taken from a more complicated chart that first appeared in the writer's A.S.M.E. paper, "Current Practice in Pressures, Speeds, Clearances and Lubrication of Oil Film Bearings." The uncircled numbers within the various areas refer to the sources of the data listed on Page 154. To these the user may add his own experience and design habits.

Across the face of *Fig.* 4 the lines from northwest to southeast are marked with the values of the product *PV* that are constant for points upon them. The lines from northeast to southwest are marked with the values of the ratio *P/V* that are constant for points upon them. It is interesting to note from Equation 1 that a uniform clearance ratio in a line of similar bearings of varying diameters would require that the viscosity increase directly as the diameter if the pressure increases directly as the velocity, for optimum capacity operation. However, that is not the direction of approach that will be made.

On the face of *Fig.* 4 are marked what may be thought of as zones of conservatism. At the northeast corner of each is a letter in a quadrant.

The letter *A* is at the corner whose coordinates are $P = 30$ psi and $V = 300$ fpm. The letter *B* lies at $P = 300$ psi and $V = 3000$ fpm, and so on up to *E* at the corner of the chart. Necessity leads the designer to higher combinations of pressure and velocity, and it would be interesting to know how far it has been necessary to go.

The examples that follow will show some of the ways these data sheets may be used for the solving of bearing problems where width equals diameter. They are lettered (a), (b), (c), and (d). On the *PV* chart the coordinates of the example data are plotted, the point being marked with the example letter.

TABLE III

Performance Factors for Various Length-Width Ratios

of 120-degree centrally loaded clearance bearings

Length-Width Ratio $\frac{l}{b}$	Leakage Correction: Friction Factor F_1	Leakage Correction: Load Factor L_1	Equation 1a $\frac{Pa}{V\mu}\left(\frac{\eta}{a}\right)^2$	Equation 2a $\frac{H}{PV}\left(\frac{a}{\eta}\right)\frac{10^5}{a}$	Equation 3a $\frac{\lambda}{\left(\frac{\eta}{a}\right)}$	Equation 4a $\frac{\lambda}{\sqrt{\frac{\mu N}{P}}}$	Bearing Width b	Bearing Area $A=bd$
2.000	0.820	0.223	0.0508	34.45	5.67	0.926	½d	½d^2
1.600	0.860	0.303	0.0692	26.6	4.38	0.833	⅝d	⅝d^2
1.333	0.886	0.372	0.0848	22.3	3.68	0.774	¾d	¾d^2
1.143	0.905	0.432	0.0985	19.6	3.235	0.735	⅞d	⅞d^2
1.000	0.920	0.481	0.1096	17.9	2.95	0.7075	d	d^2
0.800	0.938	0.558	0.1272	15.7	2.59	0.669	1¼d	1¼d^2
0.667	0.948	0.613	0.1398	14.5	2.38	0.645	1½d	1½d^2
0.500	0.963	0.689	0.1571	13.1	2.16	0.618	2d	2d^2

EXAMPLE (a): Let $P = 200$ psi, $V = 1000$ ft per min, $d = 12$ inches, $b = 12$ inches, bearing angle = 120 degrees; the bearing is centrally loaded, has running clearance. What information do these data sheets furnish about such a bearing?

SOLUTION: From *Fig.* 1, $N = V/0.262d = 1000/(0.262 \times 12) = 318$. The least film thickness, h_0, for a

projected area of 144 sq in. will be 0.001556 inch for $V = 1000$, by interpolation in TABLE I. The radial bored running clearance for optimum capacity operation will therefore be $\eta = h_0/0.531 = 0.00293$ inch, and $\eta/a = 0.00049$, a little less than half a thousandth inch per inch of radius. The total bored clearance ratio to diameter would be the same.

It follows from Equation 3 on *Fig.* 1, or from *Fig.* 3, that the friction coefficient λ will be 0.001455. From Equation 4, or from *Fig.* 3, the value of the symbol group $\mu N/P = 4.25 \times (10)^{-6}$. From this it follows that the coefficient of viscosity $\mu = 2.67 \times (10)^{-6}$ reyns (pounds-seconds per square inch). Expressed in centipoises this coefficient would be $Z = 18.4$, a metric measure. Expressed in poises it would be 0.184 (dynes-seconds per square centimeter).

Power loss of frictional drag at the surface of the journal is found from Equation 2 to be $H \times b = 1.27$ horsepower, and its equivalent in heat must be removed from the oil continuously, by one means or another, in order to maintain the assumed optimum operation. Load carried by the bearing in the foregoing conditions is $200 \times 144 = 28{,}800$ pounds.

From *Fig.* 1 it is seen that the maximum pressure in the supporting film is almost directly under the journal. It is at h_1 which is 3.14 degrees beyond the load line in the direction of rotation. The position of least film thickness is 36.3 degrees beyond the load line. The film thicknesses at h_E, h_1 and h_L can be calculated by means of the formula $h = 1 + c \cos\theta$.

Facts learned from the foregoing analysis form the basis of further calculations that lead to the selection of an oil, its operating temperatures within and outside the film, and the proper rate of oil supply.

EXAMPLE (b): Let $P = 2000$, $V = 20{,}000$, $d = 6$ inches, $b = 6$ inches, bearing angle = 120 degrees. Bearing is centrally loaded and has running clearance. This problem is wholly outside the region of experience recorded on the *PV* chart, *Fig.* 4. The solution should indicate why this *PV* combination is avoided, and what the result would be if it were encountered.

SOLUTION: $N = 12{,}730$. $h_0 = 0.00374$, found by extrapolation, for a projected area of 36 square inches. $\eta = 0.00704$. $\eta/a = 0.00235$. $\lambda = 0.007$. $\mu N/P = 98.1 \times 10^{-6}$. $ZN/P = 676$. $\mu = 15.5 \times 10^{-6}$ reyns. $Z = 107$. centipoises. The power loss $Hb = 304$ horsepower. The bending stress in outer fiber for end journal = 10,200 psi. If such a shaft were transmitting power with a maximum torsional stress of 10,000 psi the transmitted power would be 85,600 horsepower. Compared with this the bearing loss is not so important. However, as stated at the beginning, this is a hypothetical case.

EXAMPLE (c): A point in Field 8, circled in *Fig.* 4, would more nearly represent present practice but on the high side, if we choose the combination $P = 500$, $V = 5000$, and $d = 5$ inches, $b = 5$ inches, $A = 25$ square inches.

SOLUTION: $N = 3820$, $h_0 = 0.00202$, $\eta = 0.00380$, $\eta/a = 0.00152$. The coefficient of friction would be $\lambda = 0.00452$. $\mu N/P = 41.1 \times 10^{-6}$ which makes $\mu = 5.38 \times 10^{-6}$ reyns. $ZN/P = 290$ which makes $Z = 37.1$ Centipoises. The power loss for this bearing would be $Hb = 8.55$ hp.

EXAMPLE (d): The high-speed moderate-pressure region is represented by $P = 100$, $V = 15{,}700$, $d = 2$ inches for which $N = 30{,}000$.

SOLUTION: $b = 2$ inches. $A = 4$ square inches. $h_0 = 0.0022$. $\eta = 0.00413$. $\eta/a = 0.00413$. $\lambda = 0.0123$. $\mu N/P = 304 \times 10^{-6}$. $\mu = 1.013 \times 10^{-6}$ reyns. $ZN/P = 2100$. $Z = 7$ centipoises. Friction loss is $Hb = 2.34$ hp.

NOMENCLATURE

A = Projected area of journal, sq in., $= 2\,ab = db$
a = Radius of journal, inches, $= d/2$
b = Width of supporting bearing, inches measured axially
c = Factor of eccentricity of journal in the bearing
d = Diameter of journal, inches, $= 2a$
F_1 = Side leakage correction factor for friction
H = Horsepower of journal friction per unit of width b
h = Film thickness, inches, at any angular position, *Fig.* 1.
h_o = Least film thickness, inches measured on line of centers OO', *Fig.* 1
h_1 = Film thickness, inches, at region of maximum pressure p_1
h_E = Film thickness at leading edge, inches
h_L = Film thickness at trailing edge, inches
L_1 = Side leakage correction factor for load-carrying capacity
l = Length of oil film, inches
N = Revolutions per minute
P = Average pressure, psi, over projected area for journal
R = Total load of journal upon the bearing, pounds
V = Velocity of moving surface, feet per minute
W = Load capacity per unit of bearing width $= R/b$, pounds
Z = Absolute viscosity, centipoises
μ = Absolute viscosity, reyns (lb-sec per sq in.)
λ = Coefficient of friction
θ = Angle to position in film where thickness is h, *Fig.* 1
η = Radial Clearance, bored in the bearing. Bearing radius $= a + \eta$.

REFERENCES

Index of uncircled numbers of data source marked on the *PV* chart; manufacturers, reference books and periodicals.

1. Westinghouse Electric Corp.
2. Newport News Shipbuilding Co.
3. Electric Boat Co.
4. E. I. Du Pont De Nemours Co.
5. William Sellers Co.
6. General Electric Co.
7. De Laval Separator Co.
8. U. S. Navy Specification S-45-1.
9. Kingsbury Machine Works Inc.
10. Kimball and Barr—*Elements of Machine Design*, 1923.
11. F. A. Halsey—*Handbook for Machine Designers and Draftsmen*, 1913.
12. Des Ingenieurs Taschenbuch—A. V. Huette, 26th Edition, 1926.
13. *Machinery's Handbook*, 1930.
14. C. Pfleiderer—*Die Kreiselpumpen*, 1932.
15. Prescott Poole—*Journal SAE*, Oct., 1931.

MACHINE DESIGN'S

MATERIALS WORK SHEET

FILING NUMBER
15.01

Hastelloy Alloys

Wrought and Cast

AVAILABLE IN: (Alloy A) Castings, wrought parts, rolled sheet†, wire and welded tubing.
(Alloy B) Castings, wrought parts, rolled sheet†, wire and welded tubing.
(Alloy C) Castings, hot-rolled sheet† and plate†, and welded tubing.
(Alloy D) Castings.

† Furnished fully annealed unless otherwise specified.

NOMINAL ANALYSES:

	Ni	Mo	Cr	Fe	W	Cu	Mn	Si	Al
Alloy A	bal	22	..	22	..	..	*	*	..
Alloy B	bal	32	..	6	..	..	*	*	..
Alloy C	bal	19	17	6	5	..	*	*	..
Alloy D	bal	..	..	..	..	3	*	10	*

* Small percentages of these elements employed.

PROPERTIES

TENSILE STRENGTH
(average, psi)

Alloy A,	cast	69- 77,500
	rolled, annealed	110-120,000
Alloy B,	cast	75- 82,000
	rolled, annealed	130-140,000
Alloy C,	cast	72- 80,000
	rolled, annealed	115-128,000
Alloy D,	cast	36- 40,500

REDUCTION OF AREA
(average, per cent)

Alloy A,	cast	16-18
	rolled, annealed	40-54
Alloy B,	cast	10-13
	rolled, annealed	40-45
Alloy C,	cast	11-16
	rolled, annealed	....
Alloy D,	cast	....

YIELD POINT
(average, psi)

Alloy A,	cast	42-45,000
	rolled, annealed	47-52,000
Alloy B,	cast	55-57,000
	rolled, annealed	60-65,000
Alloy C,	cast	45-48,000
	rolled, annealed	55-65,000
Alloy D,	cast	....

HARDNESS
(average)

		Rockwell	Brinell
Alloy A,	cast	B85-94	155-200
	rolled, annealed	B94-97	200-215
Alloy B,	cast	B92-99	190-230
	rolled, annealed	B96-100	210-235
Alloy C,	cast	B89-97	175-215
	rolled, annealed	B84-95	160-210
Alloy D,	cast	C50-55	......

ELONGATION IN 2 INCHES
(average, per cent)

Alloy A,	cast	8-12
	rolled, annealed	40-48
Alloy B,	cast	6-9
	rolled, annealed	[illegible]5
Alloy C,	cast	[illegible]-15
	rolled, annealed	25-50
Alloy D,	cast	0

IZOD IMPACT STRENGTH
(average, ft-lb)

Alloy A,	cast	25-35
	rolled, annealed	62-77
Alloy B,	cast	11-16
	rolled, annealed	68-78
Alloy C,	cast	9-14
	rolled, annealed	34-40
Alloy D,	cast	....

MACHINE DESIGN is pleased to acknowledge the collaboration of the Haynes Stellite Company in this presentation.

MODULUS OF ELASTICITY
(psi)

Alloy A, cast and rolled	27,000,000
Alloy B, cast and rolled	30,750,000
Alloy C, cast and rolled	28,500,000
Alloy D, cast	28,850,000

TENSILE STRENGTH AT ELEVATED TEMPERATURES
(short-time tests, average, psi)

	Temperature (C)			
	500	700	900	1000
Alloy A, rolled, annealed	90,000	55,000	40,000	23,000
Alloy B, rolled, annealed	115,000	85,000	50,000	25,000
Alloy C, rolled, annealed	94,000	75,000	42,000	22,000

PHYSICAL CONSTANTS

	Alloys			
	A	B	C	D
Specific Gravity	8.80	9.24	8.94	7.80
Melting Range (C)	1300–1330	1320–1350	1270–1305	1110–1120
(F)	2372–2426	2408–2462	2318–2381	2030–2048
Thermal Conductivity cal/sq cm/cm/sec/deg C.	0.04	0.027	0.03	0.05
Btu/sq ft/in./hr/deg F	116	78.5	87	145
Specific Heat cal/deg C	0.0939	0.0907	0.0920	0.1086
Mean Coef. Thermal Expansion per deg C 0-100 C.	0.0000110	0.0000100	0.0000113	0.0000110
0-1000 C	0.0000154	0.0000146	0.0000153	0.0000181
Per deg F 32-212 F	0.0000061	0.0000056	0.0000063	0.0000061
32-1800 F	0.0000086	0.0000081	0.0000085	0.0000101
Electrical Conductivity Compared to Copper at 24 deg C (per cent)	1.40	1.31	1.33	1.56
Electrical Resistivity at 24 deg C (microhms per cm³)	126.7	135.0	133.0	113.0
Magnetic Permeability	5	4	4	....
Shrinkage of Sand Castings (in./ft)	¼	¼	¼	¼

CHARACTERISTICS

Hastelloy alloys combine outstanding resistance to the action of a wide variety of corrosive media with excellent physical properties. Alloys A, B and C are comparable in tensile properties to high-strength steels, while Alloy D possesses considerable hardness and wear resistance.

ALLOY A: A nickel-molybdenum-iron alloy which is readily machinable, makes excellent castings, and can be welded by the oxyacetylene, atomic hydrogen or metallic-arc processes. It is unusually strong at high temperatures. As it is an austenitic type alloy, it can be softened by heating to a high temperature and then quenching in water. Cold-working materially hardens this alloy, raising its yield point and ultimate strength and correspondingly reducing its ductility.

ALLOY B: A nickel-molybdenum-iron alloy similar to Alloy A but containing a higher percentage of molybdenum. Has greater mechanical strength than Alloy A, particularly at elevated temperatures. Like Alloy A, it is readily machinable.

ALLOY C: A nickel-molybdenum-chromium-iron alloy. In cast form its strength compares favorably with that of low or medium-carbon cast steel and, although not as ductile as steel, it has considerable toughness for a cast material. In the form of hot-rolled sheet, plate or bar, it is comparable in strength and ductility to alloy steels. It is machinable at moderate speeds and can be welded by the oxyacetylene, atomic hydrogen or metallic-arc method. The fact that it can be ground to an excellent finish and can be highly polished is important in applications such as valve seating surfaces and shafts or shaft sleeves which operate in packed journals.

ALLOY D: Cast alloy composed primarily of nickel and silicon, but also containing small amounts of other metallic elements. Is similar in physical properties to a high-grade gray cast iron. Because of its hardness, it is not workable and, as it can be machined only with great difficulty, it must be finished by grinding. However, it has high transverse strength and good deflection, particularly after a toughening anneal. This alloy possesses good abrasion resistance, a property that makes it particularly suitable for handling liquids containing suspended abrasive matter. It is nongalling when used with Alloys A, B or C, or austenitic steels.

APPLICATIONS

These alloys are used for parts in equipment which must withstand the action of various corrosive media and, at the same time, retain good strength. Extensive use is found in the steel, oil, coal products, paper, paint, drugs, plastics, dyes, explosives, rubber and textile industries. The following list is representative, the particular alloy to be employed depending on corrosive media encountered and mechanical properties required:

Agitator units
Autoclaves
Ball bearings
Blowers
Blowpipes
Bolts
Burner parts
Bushings
Chain links
Coils
Concentrator tubes
Condensers
Dryers
Etching equipment
Evaporators
Exhausters
Fans
Filter parts
Filter press plates and frames
Floats
Fractionating columns
Furnace parts
Heat exchangers
Impellers
Injectors
Kettles
Laboratory hoods
Liners
Manifolds
Meter parts
Mixer parts
Muffles
Nipples
Nuts
Orifice plates
Paddles
Pipe and pipe fittings
Propellers
Rolls
Scrapers
Screw conveyor flights and housings
Shafts and sleeves
Spray nozzles
Steam jets
Strainers
Tanks
Thermocouple protection tubes
Thermometer wells
Vessels

FABRICATION

MACHINABILITY:

Alloy A is readily machinable by ordinary methods. Alloys B and C can be machined at comparatively low cutting speeds. Since Alloy D is practically unmachinable, grinding generally is recommended. However, simple machining operations such as facing-off flanges can be performed where grinding is not practical. All castings of Alloys A, B and C are furnished in the condition in which they are most readily machinable.

TURNING AND BORING:

Turning, boring and facing ordinarily are done dry. All roughing operations on castings are performed with sufficient depth of cut to insure that the cutting edge of the tool stays under the scale. A heavy feed with a light cut is preferable to a light feed and deep cut. Cutting edges of roughing tools should have as much support as possible, and sufficient side rake should be ground to keep chip pressure at a minimum. Roughing tools should be ground with a side clearance of 5 to 7 degrees, and front clearance of 6 to 8 degrees. When roughing, the side rake should be 4 to 6 degrees for castings and 10 to 12 degrees for wrought material. No back rake is needed for roughing

cuts. For finishing cuts, a 10 to 12-degree side rake is satisfactory for either form of material. For finishing light work, the tool can be ground with a front clearance up to 10 degrees, with a maximum of 5 degrees back rake to give sharp cutting edges. Turning tools should be set on center or slightly below center. Boring tools should be on center or slightly above.

DRILLING:

For Alloys A, B and C, ordinary high-speed drills are used. Point of the drill should have an included angle of 135 to 140 degrees and a clearance angle of 10 degrees. The drill web should be thinned out at the point and reduced to about one-third of the web thickness of a standard drill. In drilling a deep hole it often is necessary to sharpen the drill at least once or twice during the drilling operation. For drilling small or deep holes a coolant is recommended, consisting of a mixture of one part lard oil, one part mineral oil and the usual amount of white lead. Sulphurized oils should not be used. Large shallow holes can be drilled dry.

REAMING:

Not less than 0.012-in. of stock should be removed, 0.015 to 0.020-inch being the optimum. Spiral-fluted, high-speed steel reamers are recommended, having 0.010 to 0.015-inch lands and well-polished flutes. Reamers are kept sharper than for cast iron work. Speeds approximately twice those used for drilling are recommended, using adequate lubrication.

GRINDING:

Grinding conditions should be severe enough to yield some wheel breakdown. A roughing cut of 0.005-inch, and finishing cut of 0.0025-inch are recommended. A continual stream of good soluble grinding lubricant should flow on the metal at all times. Alloys A, B and C cannot be lapped readily, so, in preference to lapping it is recommended that the surface be finished to close tolerances by grinding.

TAPPING:

For Alloys A, B and C, standard high-speed steel, four-fluted plug taps with ground threads and 7-degree spiral flutes, or four-fluted spiral-pointed plug taps are best. Only commercial ground taps can be used. Where permissible, it is advisable to use a tap-drill which is slightly oversize so that a 60 to 65 per cent full thread is obtained. A slightly longer lead is ground on taps than is commonly supplied on standard taps. When tapping a thread to the bottom of a blind hole, the hole should be tapped first with a tapered tap, next with a plug tap, and finally with a bottom tap. Tapping speeds are approximately 8 to 10 surface feet per minute. Taps must be kept sharp at all times, and lubrication with lard oil and white lead must be provided.

SHEARING:

Alloys A, B and C sheet and plate can be sheared hot or cold. However, since they are much stiffer and stronger than ordinary sheet and plate materials, particularly at elevated temperatures, more power is required to shear them. A shear that will handle a certain maximum thickness of mild-steel plate will handle only about 50 per cent of that thickness in Hastelloy-alloy plate. These alloys cannot be cut successfully by oxygen cutting methods because they possess too much resistance to oxidation. Cutting with a carbon pencil, which merely amounts to melting away the material, can be done, but it is practically impossible to hold a straight or even edge. Following the use of this method, the surface adjacent to the cut should be ground or machined to remove the thin layer which has picked up carbon, as this layer would not have maximum corrosion resistance.

FORMING:

Alloys A, B and C sheet and plate are preferably formed cold and simple forming operations such as dishing, flanging, or rolling thin sheets into cylindrical shapes can be readily performed. Cold-forming these alloys will work-harden them to some extent, but generally not enough to necessitate subsequent heat-treatment for maximum corrosion resistance. Greater power is required for simple cold-forming operations than is required for mild steel.

Hot Forming: More severe forming operations such as dishing, flanging, rolling heavy plates into small cylindrical shapes, and plain flanging, have been performed hot, but this practice generally is not recommended due to the limited working range. Hot-forming of these alloys should be attempted only where suitable furnace equipment with proper heat control is available. These alloys possess a brittle range between 1200 and 1900 F, and cracking may occur in this temperature range unless the flow of the metal is extremely slow. Hot-forming can be carried out above 1900 F and up to 2150 F for Alloys A and B, and up to 2200 F for Alloy C, as long as the flow of the metal is not too rapid. When forming between dies, care must be taken that the metal is not cooled rapidly into the brittle range by contact with cold dies before the forming is completed. For this reason it often is necessary to reheat several times during particularly severe forming operations. Extreme care is required in performing hot-forming operations to prevent overheating and to maintain a neutral atmosphere in the furnace.

Severe Flanging: These operations are apt to be troublesome and, where the design permits it is preferable to eliminate flanging operations and weld on a strip to make the flange.

Rolling: Rolling of sheets into cylindrical sections in most cases can be performed cold. Approximately twice as much power is required as for rolling the same diameter in mild steel.

Cold-drawing: Simple shapes in Alloys A, B and C can be cold-drawn. Drawing, however, will work-harden the alloys and it often is necessary to perform the operation in several steps, giving a full anneal to restore maximum ductility between operations. After completion of all forming operations, either hot or cold, it is advisable to give a full anneal or stabilizing anneal to impart maximum corrosion resistance to the sheet.

HOT FORGING:

Simple forgings can be made from Alloys A and B. Forging operations must be done hot and cannot be carried out under 1900 F without danger of cracking. Optimum forging temperature is approximately 2100 F. For upsetting bolts and for forging small parts, hot-rolled bar stock is recommended. It should be centerless ground to remove any superficial surface checks which might open up during forging. Close control of the furnace atmosphere also is required, and the muffle-type, controlled-atmosphere furnace is preferred. The atmosphere should be kept as nearly neutral as possible.

WELDING:

Alloys A, B and C can be welded readily by the oxyacetylene, metallic arc or atomic-hydrogen process. The carbon arc cannot be used because of carbon pickup and consequent lowering of corrosion resistance. Cast parts of Alloys A, B and C generally are welded by the oxyacetylene process, using a neutral flame

and a welding rod of the same composition as the base metal, with the exception that Alloy A is welded with Alloy B welding rod to give superior corrosion resistance in the welds.

Localized preheating is sufficient when welding small castings of Alloys A, B and C, but for large castings, thorough preheating is advisable to prevent heating and cooling strains incidental to the welding operations. Heat treatment of welded castings is necessary only when they have been preheated for welding and held at red heat for an appreciable period of time during welding. In such cases it would be necessary to fully anneal the casting after welding.

For large sheet-fabricated equipment, the metallic arc process usually is preferred, with reversed polarity and the arc choked very short. Overheating of the weld metal is apt to cause porosity. The atomic-hydrogen process, particularly adapted to automatic setups, produces sound welds in these alloys when precautions are taken to prevent overheating. In welding Alloy C sheet, however, the metallic-arc method is preferred because this alloy hardens alongside the weld more rapidly than either A or B.

When welding Alloy A, B and C sheet and plate, or Alloy A and B hot-rolled bar stock or forgings, it is desirable to anneal after welding. If it is not possible to anneal after welding, the sheets can be purchased in the stabilized condition, which eliminates the necessity of heat-treatment after welding. These alloys in sheet form all can be welded readily to steel using either Hastelloy alloy electrodes or electrodes of 25-12 chromium-nickel stainless steel.

Alloy D can be welded by the oxyacetylene process, using an excess acetylene flame and Alloy D welding rod. Castings to be welded should be preheated gradually to a dull red heat and maintained at this temperature during the entire welding operation. They should then be slowly and uniformly cooled to room temperature in a furnace.

RESISTANCE TO CORROSION

Alloy A: This alloy is particularly resistant to hydrochloric and sulphuric acids. It withstands the action of hydrochloric acid in all concentrations at temperatures up to 70 C (158 F), although rate of attack increases slightly with temperature. It is also resistant to sulphuric acid of any concentration up to 70 C and to all concentrations below 50 per cent up to the boiling point as well as to acetic, formic and other organic acids, but not to oxidizing agents, and is practically unaffected by alkalies.

Alloy B: Particularly well suited for equipment handling boiling hydrochloric acid and wet hydrochloric acid gas. Its resistance to sulphuric acid is better than that of alloy A and it has good resistance to phosphoric acid.

Alloy C: Effectively withstands strong oxidizing agents such as nitric acid, free chlorine, aqueous solutions containing chlorine or hypochlorites, and acid solutions of ferric or cupric salts. This alloy has practically same resistance to hydrochloric acid at room temperature as has Alloy A and is slightly more resistant to sulphuric acid. It resists phosphoric acid, is highly resistant to acetic, formic and sulphurous acids, and has excellent resistance to dry battery mix.

Alloy D: Exceptionally resistant to sulphuric acid of all concentrations up to the boiling point. Resistance to hydrochloric acid is fair at moderate temperatures. It is practically unaffected by acetic or formic acid and has low corrosion rate in phosphoric acid. It is not resistant to strong oxidizing agents.

All of these alloys are practically free from corrosion under such conditions as exposure to the atmosphere, fresh or salt water, and neutral or alkaline salts.

ANNEALING

Full Anneal: Produces softness, toughness, machinability, and high corrosion resistance, and is the anneal generally used on welded sheet to prevent deterioration in the area directly adjacent to the weld. Also given to castings to improve machinability:

	Alloys A and B	Alloy C
Temperature	2100-2150 F (1150-1180 C)	2200-2225 F (1205-1220 C)
Time	20 min. (1/8-in. sheet) to 90 min. (2-in. plate or bar)	2 to 3 hours according to mass
Cooling	Rapidly in air (or water for heavy sections)	Rapidly in air

Stabilizing Anneal: Used to reduce the loss of corrosion resistance and toughness due to exposure of the metal to elevated temperatures between 1200 and 1900 F (650 to 1040 C). Also helps to reduce sheet deterioration adjacent to welds when applied to the sheet before welding, where treatment after welding is not practicable. On Alloy A and B wrought material it results in corrosion resistance nearly as good as that obtained by the full anneal. Stabilization is most effective when preceded by the full annealing treatment:

	Alloys A and B	Alloy C
Temperature	1925-1950 F (1050-1065 C)	2050-2075 F (1120-1135 C)
Time	2 to 4 hours	2 to 4 hours
Cooling	In air	In air

Hardening Anneal: Produces hardest possible condition (except by rolling not followed by annealing) in Alloys B and C. Results in considerable reduction of corrosion resistance, but is used for wear resistance and resistance to galling and seizing. An age-hardening process which must be preceded by the full anneal and which requires a long time to produce maximum hardness. Hardness to rockwell C40 to 50 in Alloy B and rockwell C-35 to 40 in Alloy C can be obtained:

	Alloy B (cast or wrought)	Alloy C (cast)
Temperature	1375-1400 F (745-760 C)	1575-1600 F (860-870 C)
Time	One week	16 hours
Cooling	In air	In air

	Alloy C (sheet or plate)
Temperature	1575-1600 F (860-870 C)
Time	8 hours
Cooling	In air

Toughening Anneal: The only treatment given to Alloy D. It results in considerable reduction in the brittleness of the cast material and also lowers hardness of the alloy to rockwell C-40 to 45 without affecting its corrosion resistance:

	Alloy D
Temperature	1925-1950 F (1050-1065 C)
Time	2 to 4 hours (according to mass)
Heating and cooling	Slowly in furnace

Noteworthy Patents

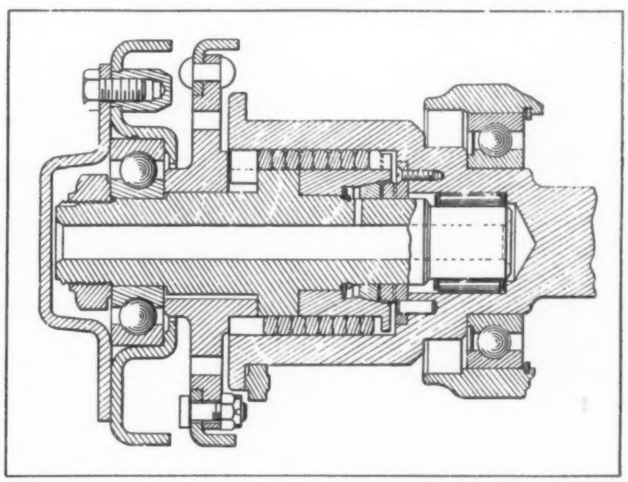
GEAR BACKLASH IS OBVIATED with the arrangement covered in patent 2,397,777 assigned to the Barber-Colman Co. Performing either as an idler gear or as part of a reduction, a floating gear is wedged into mesh with driving and driven gears by means of spring action. Pressure angles on the order of 30 degrees are used to prevent locking of the gear teeth.

ELIMINATION OF POWER LOSS which now normally accompanies the steering of tractors, tanks and other heavy vehicles is achieved by a novel transmission device described in patent 2,397,140. Installed between each wheel and the differential, circumferentially-arranged pairs of pistons utilized in the device are locked hydraulically to create direct drive or relieved to allow free reciprocation and slippage according to the movements of the steering wheel. Instead of braking a continuously powered tread, it is relieved from the power drive, the opposite tread absorbing full power to make the turn. An undulating cam track traversed by the piston rollers of the transmission during the relief cycle, provides the driving engagement or clutching for direct drive. Patent assigned to Hickman Pneumatic Seat Co. by Albert F. Hickman.

AXIAL DISPLACEMENT as well as circumferential movement between a hub and a shaft is prevented by means of a novel spline or key lock covered in patent 2,401,536. Assigned to the American Pulley Co. by W. A. Williams, the spline is so designed that by turning an appropriate threaded collar piece which engages the spline, it can be wedged between the hub and shaft to lock the two or when necessary, release them.

REGULATION OF SPEED of internal combustion engines without limiting load-pulling ability is accomplished by a governor device covered in patent 2,401,588 assigned to Timken Roller Bearing Co. by H. C. Edwards. A centrifugal speed regulating mechanism automatically controls the output of the fuel injection pump and is specially suited for diesel engines. Maximum quantity of fuel delivered is limited at all high-speed adjustments of the throttle lever but, under full load operation of the motor, the governing mechanism permits additional fuel feed to provide increased torque sufficient to carry momentary overloads.

ONE WAY CLUTCHING action at predetermined speed conditions is created automatically by the coil spring coupling clutch assigned to Clark Equipment Co. by Robert E. Burrus under patent 2,396,985. When the speed

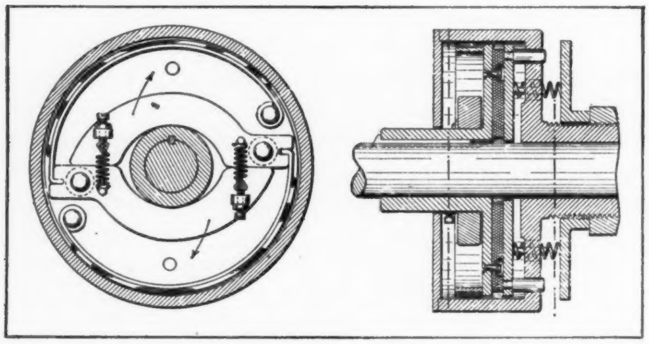


of the drive element of the coupling exceeds that of the driven, cam projections on the driven element engage a tapered sleeve, expanding the coil spring to produce a positive gripping or clutching action for direct drive. As the speed ratio reverses automatic disengagement of the driven element takes place.

CONVENTIONAL SPRINGING for unpowered vehicle wheels is eliminated by a new suspension design outlined in patent 2,401,766. A differential or compensating arrangement serves to divide the load equally between the

wheels and cushions any shocks transmitted to the vehicle. Totally enclosed, the entire compensating mechanism is restrained from rotating by either a short leaf or coil spring. Assigned to Larison Compensating Axle Corp. by G. L. Larison.

GRADUAL APPLICATION and acceleration of load until full speed is attained is accomplished by a novel self-acting clutch mechanism covered in patent 2,401,981. Power applied to the clutch is transmitted to the driven



load first by means of a plate slip clutch. As the load is gradually brought up to speed, a secondary centrifugal clutch is automatically actuated to take over and carry the direct load. Patent assigned to Metal Textile Corp. by Charles Springhorn.

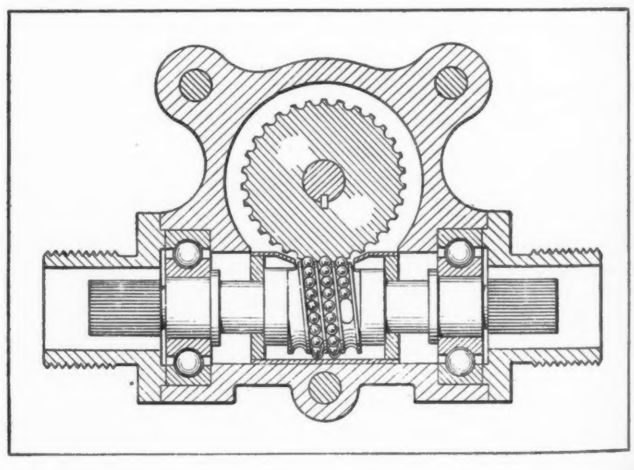
FAN EFFECT for impelling cooling air outwardly between annular core plates is produced by a novel electromagnetic rotor design for induction clutches, brakes or dynamometers. Pole pieces, extending from opposite sides of the core, are folded over the rotor windings so that alternate tapered lugs are equally spaced apart to create a radial air movement. Patent covering the device is assigned to General Electric by D. C. Prince.

SIMPLIFIED DESIGN and manufacture in connection with transmissions and similar units can be obtained by means of an unusual method of applying and retaining a thrust collar. Covered in patent 2,397,905 assigned to International Harvester Co., by Russel D. Acton and Alfred Krieg, a novel combination key holds the splined thrust collar in offset position effecting with the ends of the male splines virtual thrust shoulders for supporting an axial load.

GOVERNING CLUTCH and braking mechanisms, assigned to Metal Textile Corp. by Charles Springhorn, controls the speed of winding reels or spools by the tension of the material being wound. Both actuating and controlling the reels, the governing arrangement prevents undue acceleration, overrunning or other undesired movement of the reel likely to result in breakage. Wire, yarn, thread etc., can be handled by means of the automatic lever system utilized in the device. Details of the mechanism are covered under patent 2,401,982.

UNDESIRABLE ARCING is avoided by a novel design selector switch covered in patent 2,399,165 assigned to the Cleveland Automatic Machine Co. by G. B. Carson. As many as a dozen timed electric circuits can be actuated by means of this selector switch. In operation the brush arm, placed under spring load as the cycle is actuated, snaps rapidly to the next position as the timing cam releases its detent. Both simple and sturdy, the mechanism performs irrespective of speed or speed variations in the operating cycle of a machine and can be easily reversed as desired.

ADVANTAGES OF WORM GEAR drive—high output torque with a relatively low input torque—plus the high efficiency afforded by antifriction ball bearings is obtained



with a novel linear actuating mechanism assigned under patent 2,403,096 to Lear, Inc. by Fred M. Slavic. The worm utilizes an internal return path for the balls and an external housing to make the path of travel continuous. Mating worm wheel, of course, is toothed with a special cutter to suit the ball size. Simple in design, the unit is easily assembled from a relatively small number of components.

Printed copies of all patents covered in this department are available from the Commissioner of Patents, Washington 25, D. C. Cost of each copy is now twenty-five cents.

new parts and materials

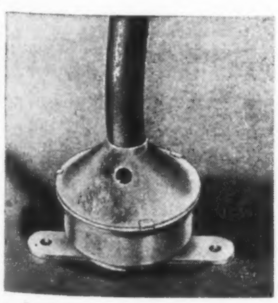
Three-Way Valve

NONMETALLIC three-way valve employing a flexible molded synthetic-rubber tube has been announced by the Grove Regulator Co., 65th & Hollis Sts., Oakland 8, Calif. The new valve is designed for handling all types of fluids, including gases, chemicals and liquids. It is said to be especially suitable for hydraulic or pneumatic cylinder operation. Nonmetallic construction permits the handling of highly corrosive or erosive liquids or gases. The unit is most satisfactory for controlling viscous or solid-carrying fluids. Bakelite encased, the valve may be manually operated by means of an over-center cam which opens and closes ports at each half turn of the handwheel. A special self-locking feature assures positive tight shut-off over extended periods of time. Available in ⅛ and ¼ in. pipe sizes, the new type valves are furnished for working pressures up to 250 psi, maximum temperature 150 F.

Spring Lock Washers

ANNOUNCEMENT of a complete line of spring lock washers has been made by the Reliance Division of Eaton Mfg. Co., Massillion, O. The washers are produced in compliance with the standards adopted by the American Standards Association, and are guaranteed to meet ASA and SAE specifications. The new line is to be distinguished from the former SAE Standard product.

Snap-Action Thermostat



HERMATICALLY-SEALED snap-action thermostat identified as C-4910, useful for thermal regulation for high limit alarms in refrigeration or other cooling or heating applications, has been announced by the Spencer Thermostat Co., Attleboro, Mass. The units are actuated by a Spencer snap-acting disk mounted in thermal contact with the sealed housing. They will operate in any position and are unaffected by ordinary vibration. Units are supplied ready to install with a heavy-duty, all rubber, two-conductor cord 30 inches long. Minimum temperature setting is −10 F and maximum setting 120 F. Minimum cooling differential is 8 F and minimum heating differential 10 F. Higher settings and wider differentials can be specified. Thermostats are available in the following electrical ratings: 10 amp in the 115-230 volts range ac; 1 amp 125 volts dc.

New Plastics

RESIN known as BCM has recently been announced by the E. I. du Pont de Nemours & Co., Wilmington, Del. A liquid bonding resin, it has been used chiefly in glass fabric laminates. However, it is also applicable to other fibers and requires only a short curing period and low laminating pressure. Odorless and resistant to most organic materials, its absorption of water and other common solvents is very low.

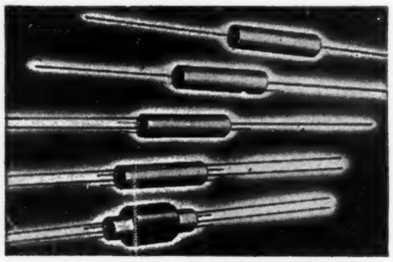
Vibration Mount

THREE NEW vibro-insulators have been added to the line manufactured by the B. F. Goodrich Co., Akron, Ohio. Recommended for use as feet or bumpers for portable machinery, blowers, fans, etc., the insulators are of 45 durometer hardness rubber with a molded in stud. Type 130 carries maximum recommended load of 132 pounds with the maximum deflection at that load of 3/16-inch and a minimum disturbing frequency 1200 per minute. Type 133 has a maximum recommended load of 180 pounds deflection is 5/32-inch and the minimum disturbing frequency 1200 per minute. Type 144 is recommended for a maximum load of 60 pounds, maximum deflection at that load 1/8-inch and minimum disturbing frequency 1350 per minute.

Coil Forms

MOLDED-BAKELITE coils forms with anchored "hairpin" wire leads have recently been announced by the Electronic Components Division of the Stackpole Carbon Co., St. Marys, Pa. Forms being smooth, coils may be wound on separate tubes and slipped over the forms, or the winding may be done on the forms direct. Standard types include forms with coaxial leads at each end, single hairpin-lead at each end, single hairpin-lead at one end with

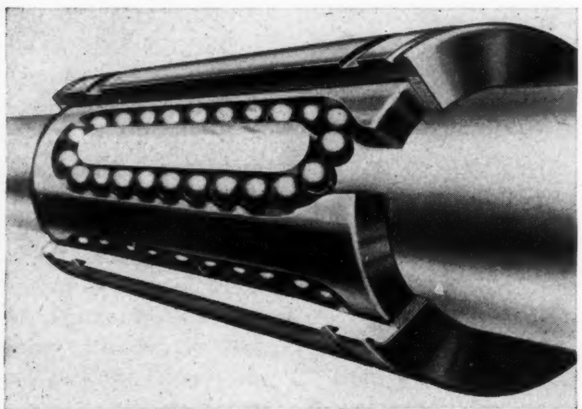
double hairpin-lead the other, and double hairpin-lead at each end. Forms are available with molded-iron cores increasing Q materially, decreasing the amount of wire re-



quired for a given inductance, and reducing stray magnetic fields. Uses range from universal and tapped windings to solenoid windings, iron-cored transformer or coupled coils.

Ball Bushing

BALL BUSHINGS for use on round shafts have been announced by Thomson Industries, Inc., 29-05 Review Ave., Long Island City. Axial as well as rotary motion is permitted by the use of circulating balls in the walls

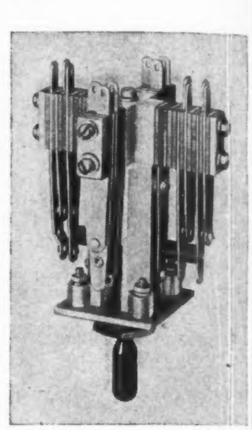


of the bushings. These bushings are said to reduce the length-to-diameter ratios necessary to prevent cocking and binding with a consequent reduction in weight. The units are available in sizes to fit shafts ranging from ¼ to 4 in. in diameter, in 17 steps. Square shaft bushings can be furnished on an experimental basis.

New Plastics

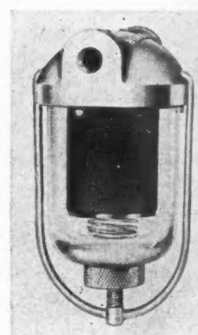
TWO NEW PLASTICS have been announced by West Coast Plastics Enterprises, No. 67, 4891 Huntington Drive, Los Angeles 32. The first, known as Plastiflex, is a rubbery vinyl-base mold material having sufficient strength to produce accurate casts in plaster and other material. The second, known as Ivoricast, is a phenol-formaldehyde casting resin having the appearance and many of the properties of ivory. Plastiflex is a nonvulcanizable thermoplastic which can be reclaimed by melting, and requires no wax or other parting agents in order to produce smooth casts. Materials having melting or curing temperatures not exceeding 230 F may be cast in plastiflex molds. Ivoricast has a compressive strength of about 1000 psi and machines like wood. It is flame resistant when asbestos filled, charring only at temperatures exceeding 400 F. Compressive strength of the order of 20,000 psi are obtainable.

Lever Switch



CAM-LEVER SWITCH having 5 positions and designed especially for ease in assembly and wiring has recently been announced. The unit known as Model MCF is locking and nonlocking in all positions except center which is always locking. Motion of the switch from center to all switching positions is straight line with resulting ease of operation. Single bolt type of mounting makes for ease of installation and wiring. The unit is especially designed for applications where it is imperative that only the desired circuit of several is closed. Manufacturer is General Control Co., 1200 Soldiers Field Road, Boston 34.

Gasoline Purifier



PURIFIER, FUNCTIONING by passing fluid between layers of helically wound ribbons of impervious material, has been announced by Skinner Purifiers, Inc., Detroit 11. Dirt and other impurities are stopped at the outer edges of these ribbons and fall into the sediment chamber. Any accumulations on the outer surface of the cylinder of ribbons occurring after long use can be cleaned quickly with compressed air. The cylindrical element is known as the Skinner Kwick Klean Kartridge. The Skinner purifier is said to separate free water from gasoline. This is achieved by the unique laminated construction of the cartridge which has orifices only a few microns in size, difference in surface tension and specific gravity of the two liquids effects the separation.

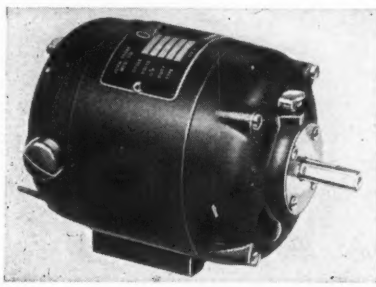
Sight Feed-Valve

ANNOUNCED BY Oil-Rite Corp., 3476 S. 13th St., Milwaukee 7, a new sight feed valve for oil circulating systems is said to be exceptionally sturdy and to have a novel way of adjusting flow. The valves are recommended for use wherever oil-flow rate must be visually checked and regulated. They are said to be adapted to circulating oil systems or multiple oiler arrangements for feeding a

number of bearings from a central reservoir. Made of sturdy hexagonal stock with large viewing windows of plastic or glass, the unit includes a drip nozzle which screws into the sight feed housing and contains the regulating screw. Oil flowing through the valve nozzle enters the sight chamber where flow can be observed. Adjustments may be made within a wide range by combination of set screw and hollow lock screw. A wide variation in control is thus possible.

Fractional-Horsepower Motor

HIGH-SPEED Ostermotor, Type EU-450 has been announced by John Oster Manufacturing Co., Racine, Wis. The unit is a 1/15-hp series-wound continuous-duty motor having high starting torques and varying speed characteristics.

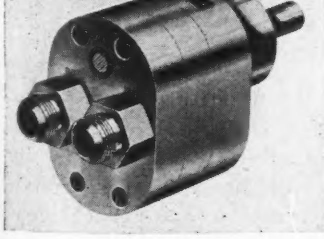


It operates at 4000 to 8000 rpm full-load speed on both alternating and direct current. The dimensions are 4½-in. long and 3 21/64-in. in diameter, weight is 3 lb 13 oz. Motor is particularly applicable to engraving machines, dental equipment, blowers, mixers and small tools for general low-power applications.

Plastic Coating

TOUGH, water-resistant coating, called Plasti-Glaze, applicable to a variety of base materials has been announced recently by the Plasti-Glaze Co., Box 168, Ventura, Calif. A nonchipping, nonpeeling coating, plastiglaze fuses with the base material and provides a pleasing, protective finish. It can be used with such materials as wood, clay, papier-mache, etc., and serves to seal and protect as well as enhance the appearance.

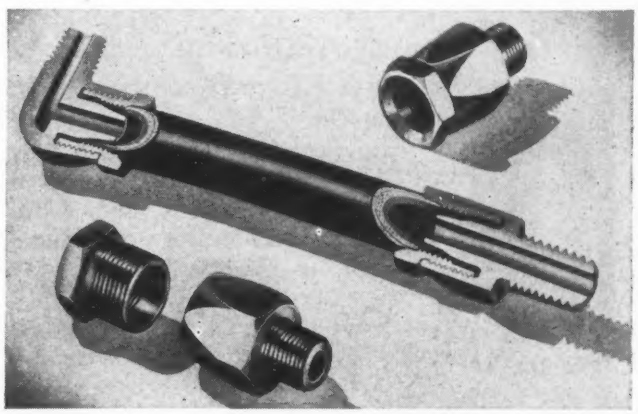
High Pressure Pump



DESIGNED for high pressure application on machines such as power transmissions, machine tools, proportioning equipment, etc., the Series 700 pump has been announced by the McIntyre Co., 719 Riverdale Ave., Newton 58, Mass. The direct-drive spur-gear pump is a small six-pound unit operating with a volumetric efficiency of over 90 per cent and a mechanical efficiency exceeding 80 per cent. Four models are available having a displacement ranging 0.4-gpm to 9.6 gpm at speeds ranging to 1750 rpm against working pressures to 1000 psi. The pump, which is constructed of aluminum with nitrided nitralloy gears, can be furnished for flange, belt, spline and other drives and requires up to 6½ hp. Positive sealing is accomplished with Linear Packing double U-cup seals relieved to suction side of the pump so that excessive seal pressures are not experienced.

Reusable Hose Couplings

ANNOUNCEMENT has recently been made by the Resistoflex Corp., of Belleville, N. J. of a complete line of detachable, reusable couplings which make it possible to hand-assemble flexible hose lines. These two-piece safety-seal couplings were developed for use with flexible



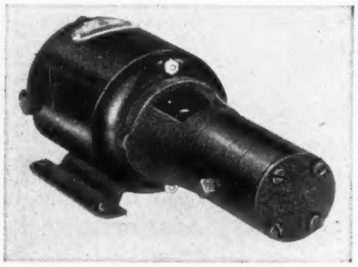
hose and, in particular, gas-oil hose. Units are quickly and easily attached or detached and can be used over and over again by end-wrench assembly on the spot. Fitting has a long even grip on the hose, the double bell shape of nut allowing the hose to flex without cutting. Heavy body of the fitting prevents crushing the line.

Process Timer

VERSATILE TIMER specially applicable to molding, electronic heating, plating, photography, etc., has been announced by Potter & Brumfield Sales Co., Dept. 232, 549 West Washington Blvd., Chicago 6. Type TA unit is powered with self-starting synchronous motor actuating a pair of heavy silver contacts rated at 1000 watts. Case is heavy molded bakelite with holes in back plate for screw mounting on any surface. Timer operates equally well in any position. Dial is fitted with an adjustable mechanism so that any desired time interval up

to the limit of the unit may be preset, and thereafter the same cycle will be repeated at each manual resetting, or stop may be released and desired interval set each time. Standard time cycles are from 0-15 minutes, 0-30 minutes, 0-1 hour, and 0-12 hours. Other time cycles are available on request. Unit measures 5½ by 3¼ by 3½ in.

Vane Type Pump



ANNOUNCEMENT of vane type pump which appeared in this section in the September issue was, unfortunately, in error. The pump, known as type VW-1, is manufactured by the Eastern Engineering Co., 294 Elm St., New Haven, Conn.

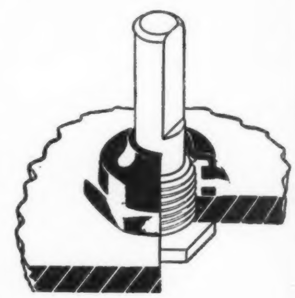
Photoelectric Control

PHOTOSWITCH Series 20 and 21 photoelectric controls recommended for such applications as counting, inspection, stop-motion control etc., are announced by Photoswitch Inc., 77 Broadway, Cambridge 42, Mass. The units are so designed that the phototube is available either integral with the housing or in separate housing. The latter design permits use of control where space is limited. A tamper-proof sensitivity adjustment is provided on the control housing to permit positive operation over varying distances between phototubes and light source. The unit is designed for high-speed operation with a relay operating at 1/20-sec, and has an operating range of 20 ft. The control is powered by a 115 or 220 volts and has an output of 10 amp at 115 volts, 5 amp at 220 volts. Weight is 11 lb.

Servomechanism

ELECTRIC servomechanism Servo Model 61 A recently announced by the W. C. Robinette Co., 802 Fair Oaks Ave., South Pasadena, Calif., is a packaged continuous balance control system. This unit controls speed and direction of a standard 1/15-hp induction motor according to the setting of the input-dial and the power may be used to control other larger power sources. Its input-dial controls either velocity or position of the rotor to very accurate limits. Velocity in either direction may be limited from zero to maximum independent of the input-dial setting so that hunting instability can never occur. The motor may actuate any device or mechanism within the range of 30-75 lb-in. torque, 0-29 rpm, controlling the process, state or condition measured on the input-dial. Alternately, the motor may act as a torque amplifier or remote positioning agent. Since the motor velocity and not the torque is controlled, load change or complete loss of load cannot cause instability. This Motron unit is designed for short time-constant systems and is usually capable of great sensitivity. Electronic components are placed in a replaceable plug-in can.

Sealnut



NEW MOUNTING and sealing nut which prevents dirt, water or gas from entering equipment panels around switch and control shafts is announced by Radio Frequency Laboratories, Inc., Boonton, N. J. An elastic sleeve tightly grips the protruding shaft or handle, and base of nut is sealed to panel by an internal rubber ring. Outstanding applications of the nut include pressurized and moisture-proof equipment used submerged or at high altitudes. It has indoor uses for protecting equipment from dust and corrosive fumes.

Hydraulic Power Unit

RECENTLY announced by the B. F. Perkins & Son, Holyoke, Mass. is a power unit consisting of high pressure hydraulic pump, electric motor, relief valve, and oil reservoir. Pump is of the constant displacement, piston type, and is available in two capacities, 1.25 gpm at 3000 psi, and 0.50 gpm at 10,000 psi for continuous operation. Intermediate pressures are obtained by adjustment of the relief valve. Upper part of the housing is formed by the oil reservoir which is furnished with cast aluminum cover. Lower housing is provided with removable side plates having air louvres. The unit has overall dimensions of 31 in. in length, 21 in. in height and 16 in. in width.

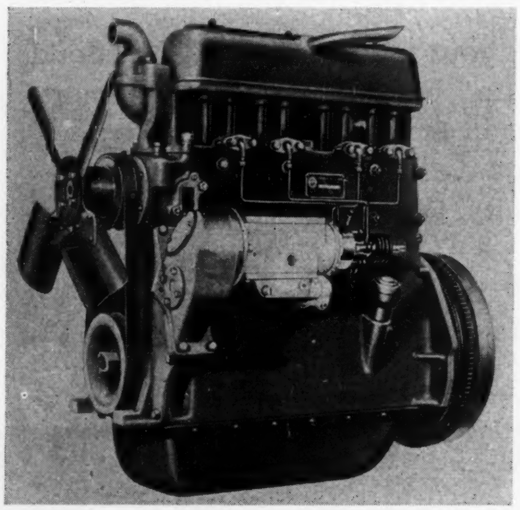
Thirty-Degree Globe Valve

GROVE SERIES V and VL valves are said to represent a marked advancement in valve body design. Thirty-degree valve body in globe pattern effects minimum change in the direction of flow through the valve and, in addition, turbulence is said to be almost entirely eliminated. Solid one-piece stem and plug is

employed to effect absolutely tight shut-off. Since the valve stem is not directly carried by the hand-wheel, any chance of the stem becoming distorted or bent is obviated. Shocks are absorbed by a multiple full-threaded section of handwheel and yoke. The new design, in thirty-degree globe and 90-degree pattern, is provided in Fullstream quick opening shut-off valves, Microstream throttling valves, Coldstream shut-off valves, and internal and external check valves. Units are manufactured by the Grove Regulator Co., 65th & Hollis Sts., Oakland, Calif.

Diesel Engine

LINE OF diesel engines rated from 25 to 150 hp and applicable in the industrial and transportation field has been announced by Continental Motors Corp., 12601 E. Jefferson Ave., Detroit 14. Outstanding characteristic of the diesel lies in the new type of combustion chamber having what is known as a "cushioned power cell". Fuel



sprayed by the injection nozzle ignites in the combustion chamber, while a portion of it carries to the "Dyna-cell" and is retarded in its burning. The effect is to retard the burning of the fuel so that the piston may have time to move downward and increase volume in the cylinder in which the charge is to burn. The net result is a low working pressure in the combustion cylinder with a simple type of nozzle. Fuels between 40-60 cetane are used. Four and six cylinder models are available.

High Density Alloy

NEW ALLOY featuring high strength and high density, identified as Alloy No. 112, has been announced by the Callite Tungsten Corp., 544 39th St., Union City, New Jersey. Produced by powder metallurgy, the alloy is available in round, square and special shapes. In certain instances unusual shapes can be made from specification. Weighing 0.620 pounds per cubic inch as compared to 0.41 pounds per cubic inch for lead, the alloy may be brazed or soldered by standard commercial methods. Alloys are available in three types each having tungsten as a basic ingredient. Type Y adds cobalt and nickel, Type E adds cobalt and silver and Type P adds cobalt and nickel. Other ingredients are also included in the alloy.

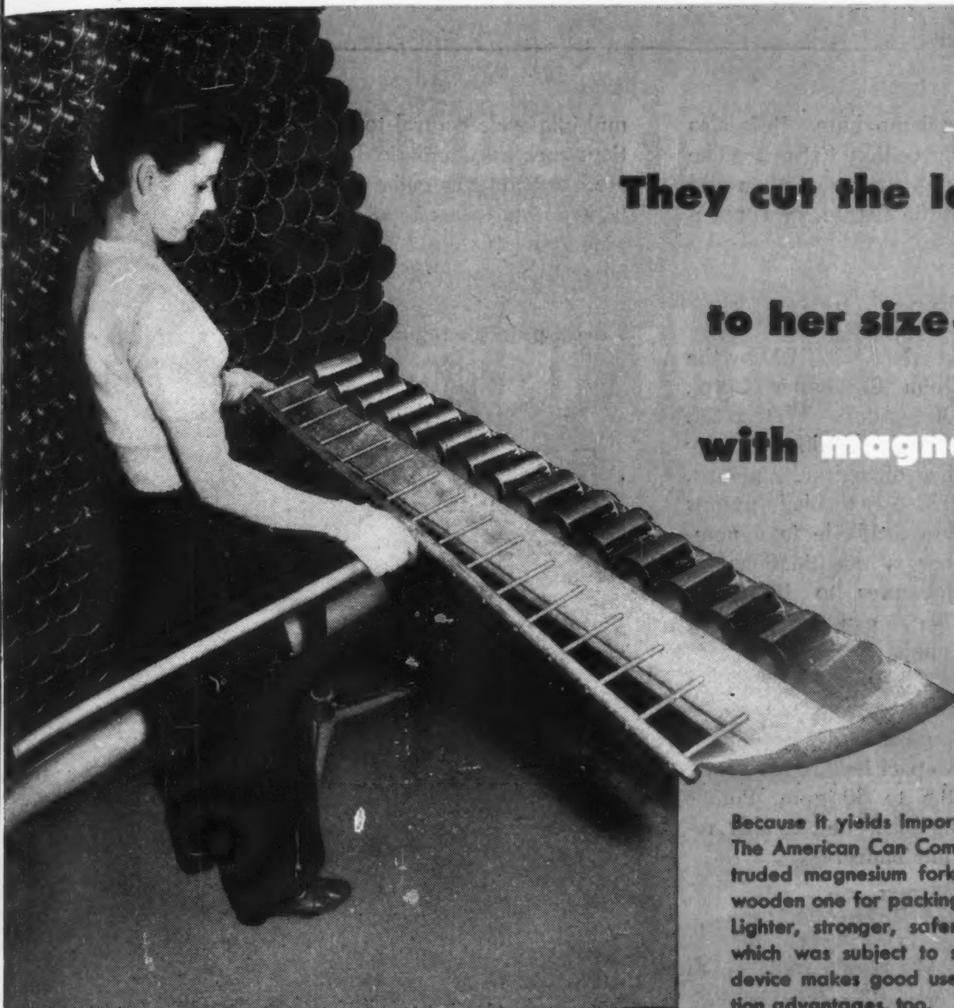
Electronic Timer

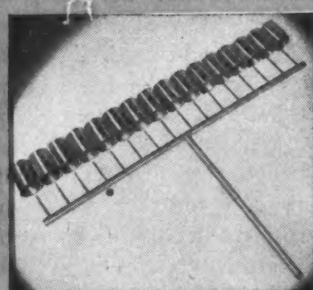
FEATURING highly accurate control of industrial processes under virtually any condition of temperature and humidity are the new Promatic timers announced by General Control Co., 1200 Soldiers Field Road, Boston 34. These new timers allow either fully automatic or semiautomatic control of the equipment to which they are connected. Two or more timers may by used to control a number of individually timed operations in a predetermined sequence. Control of the timing period is achieved by means of a plug-in type condenser unit and a variable resistor, the resistor control having a graduated dial. One condenser unit is supplied with each timer to obtain the timing period specified. Five timing periods are available in ranges of ½-cycle in 1.2 sec, to 3 cycles in 60 sec. The timers are available in both 110 and 220-volt types, both operating on 60-cycle current. Relay contacts are rated at 10 amp.

Fluorescent-Lamp Ballast

DISK-SHAPED, high power-factor ballast for the operation of two 12-inch Circline lamps has been announced by the Specialty Transformer Division of the General Electric Co. The unit, known as Tulamp, is said prac-

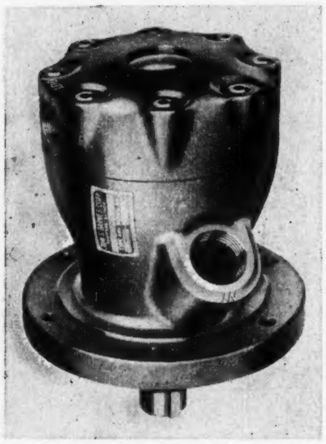
tically to eliminate cyclic light variation because the lamps are operated out of phase with each other. Flat disk construction of the ballast, with a center hole for mounting, makes it easy to assemble on the stem of a





portable lamp or to conceal in a lamp base. It is also adaptable to shallow-type wall or ceiling fixtures. The ballast leads are brought out through the cover plate and are threaded through the stem of the fixture for connection to the lamp.

Hydraulic Pump



ANNOUNCED by the John S. Barnes Corp., 301 South Water St., Rockford, Ill., is a new line of hydraulic pumps. These Roto-Blade pumps are available in a number of combinations for pressures to 1500 psi. They can be had as single or double pumps with or without feed pump, and as single pumps with gear pumps. Capacities range from 8.5 to 50 gpm. Pumps have features of equalized bearing loads and readily removable impellers for ease of maintainance.

Ultra High Speed Relay

CAPABLE of speeds up to 1000 operations per second, the Millisec relay offered by Stevens-Arnold Co. Inc., 22 Elkins St., South Boston, Mass., is a hermetically-sealed sensitive relay having a rating of 115-volt ac, 1 amp. The basic design of the relay makes possible the ultra high operating speed and at the same time assures reliability of operation in the usual speed ranges. Sensitivity down to ½ milliwatt is possible, ratings up to 5 amperes can be obtained, and closing time can be less than one millisecond.

Pneumatic Controller

ELECTRO-PNEUMATIC type controller for use with air cylinders in production processes has been announced by Electro Air Devices Co., 2811 W. Fletcher St., Chicago 18. The unit can be furnished to provide single or multiple-cycle control to perform a wide variety of operations such as automatic drilling, tapping, milling, indexing, etc. Advantages claimed for the control are: more compact design, absence of close fitting ports or valves, low potential (24 v) reducing accident hazard, accurate rapid control of any push or pull motion. The basic control unit consists of a step-down transformer, enbloc solenoid valves, pressure regulator, needle valves and armature.

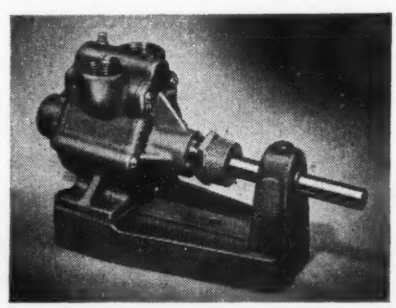
Panel Type Tachometer

ELECTRIC TACHOMETER for permanent installation has recently been announced by Ideal Industries, Inc., 1059 Park Ave., Sycamore, Ill. The unit is said to be designed particularly for use where the pick-up unit must be mounted permanently on a machine, and the meter mounted on a panel removed from the generator position. The generator consists of a small permanent Alnico-

magnet rotor mounted on precision sealed ball bearings and is capable of continuous operation at any speed within the limit of the meter. Indicating instrument is a rectifier type galvonometer capable of withstanding a momentary overload equal to 4 times maximum speed indication. Meters are rectangular in shape and available in two sizes: 3 in. and 7 in.

Gearless Pump

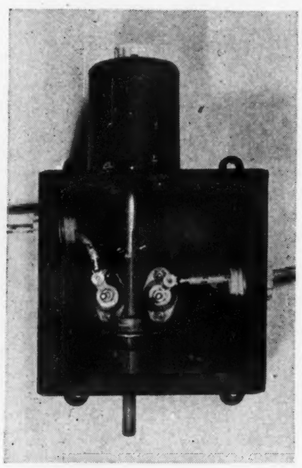
DESIGNED to eliminate common causes of pump failure—gear jamming and stripping—a new double-impeller gearless pump has been developed for the circulation of water, light oil, and other liquids by Eco Engineering Co.,



Dept. 69, 12 New York Ave., Newark, N. J. The new one-inch pump employs double impellers made of several layers of a pressure-vulcanized laminated material, and pumps equally well in either direction. Built-in driveshaft

bearing and base eliminates side-pull when pump is powered by a belt-drive and pulley, and furnishes in addition a solid base for pump mounting. The impellers are said to pass sand, grit, filings, or sludge without stalling, jamming or stripping and without damage to the bronze pump body. The unit is made with a standard one-inch connection and a built-in external driveshaft bearing and base.

Disconnect Switch



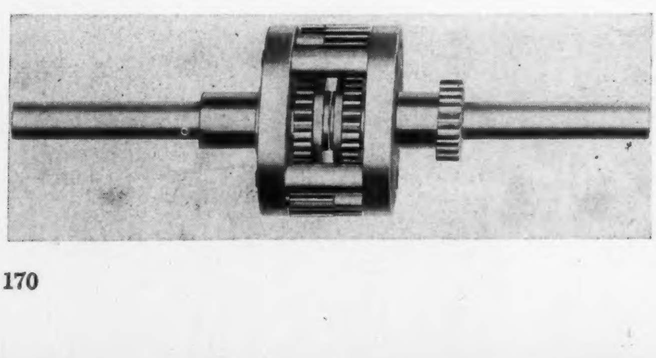
SOLENOID - OPERATED high-voltage disconnect switch operating on 115 to 230 volts ac has recently been announced by the Simco Co., 4929 York Rd., Philadelphia 41. Used for switching high voltage circuits, the unit functions by means of a brass disk shunting current across two posts. At the disconnect position the air gap totals 1½ in. Box containing the switch measures 6 by 6 by 4½ in. Switch will open a line carrying 100 mils at 10,000 volts 60 cycle ac.

Air Hose Coupling

QUICK connect air hose couplings guaranteed leakproof under working pressures up to 1000 lb are announced by the E. B. Wiggins Oil Tool Co. Inc., 3424 E. Olympic Blvd., Los Angeles 23. Feature of the coupling is its greater ease of service. Gasket change requires only a minute, reducing maintenance time and keeping production delays to a minimum. Small screwdriver is all that is needed. Coupling body does not have to be disassembled but remains intact while the gasket is removed and replaced. Coupling is disconnected by a quick pull-down and push-up on knurled ring. The new coupling will handle oxygen, oil, aromatic fuel and kerosene. Made of solid aluminum bar stock, the unit is available with the choice of two nipples—¼-in. male or female threads.

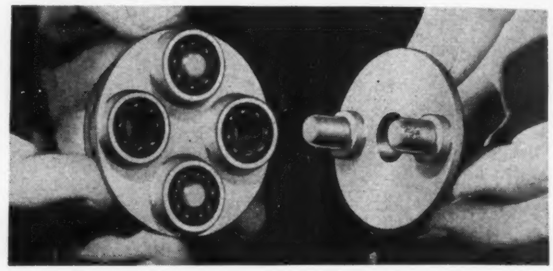
Differential

MIDGET MECHANICAL DIFFERENTIAL weighing less than 5 oz is announced by the Milwaukee Lock & Mfg. Co., Milwaukee 4. The new unit made of steel and bronze with precision cut spur gears and burnished bearings is designed to provide simple, compact, accurate method of mechanical power transfer. Differential is especially adapted for use in clutches for power transfer and automatic sequence operations.

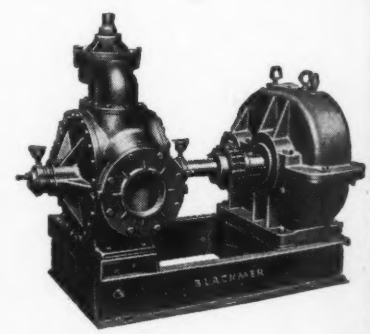


Flexible Coupling

SMALL FLEXIBLE COUPLINGS for low torques, known as "Morflex", are available in two sizes handling up to 3 or 9 pound-foot torque, and weigh ½-lb and 1¼ lb, respectively. The units are good for 3600 rpm maximum. The couplings, which are manufactured by Morse Chain Co., Detroit, Mich., utilize flexible neoprene biscuit assemblies, shielding machine installations from shock and permitting slight angular misalignment of shafts. The Morflex coupling is unaffected by oil, dirt or weather, and requires no lubrication.



Pump Unit



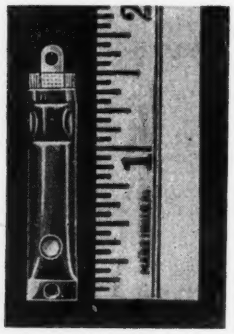
HEAVY - DUTY pumping unit with integrally - mounted reduction gear drive has been announced by the Blackmer Pump Co., Grand Rapids, Mich. The new pump is designed on the swinging - vane principle making it self - adjusting for wear. It has a capacity of 750 gpm at 150 psi discharge pressure, and operates at 225 rpm. Internal construction is bronze-fitted, units are furnished with either antifriction or sleeve bearings. The base is structural steel and provides for pump and reduction drive unit only.

New Plastic

NEW PLASTIC material offering characteristics of advantage in the manufacture of high frequency electrical equipment has been announced by the Plastics Division of General Electric Company. The material, known as Textolite No. 1422, possesses unusual heat resistance and the ability to be machined to close tolerances. It is particularly well adapted for use as an insulator in electrical connectors and possesses a very low power factor with

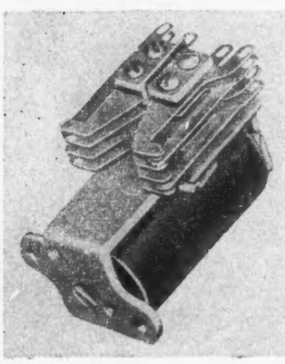
an ASTM heat distortion of 105 to 113 C. A cast plastic applicable to small quantity production, this material has a specific gravity of 1.045, tensile strength of 8000 to 10,000 psi, impact strength of 0.025 to 0.35 ft-lb, and a rockwell hardness of 68 to 72 H-scale. While unaffected by mineral acids, alkalies, alcohols or mineral oil, it is susceptible to aromatic hydrocarbons and chlorinated hydrocarbons, and ketones cause slight swelling. Water absorption is less than 0.05 per cent.

Miniature Thermostat



SMALL THERMOSTATS that may be molded into plastic material and which are dust and waterproof have been recently developed. The units known as the "Mighty Mite" are available in standard models preset to temperatures up to 300 F while other models are available for settings up to 600 F. The thermostat has ⅛-in. diameter silver contacts and a copper jacket having high thermal conductivity. Tubular construction is said to withstand great external pressure and unit will neither bend nor break under moderate loading. Standard model is designed for use up to 300 watts, 110 volts, or 150 watts 220 volts. Manufacturer is Mechanical Industries Production Co., 217-223 Ash St., Akron 3.

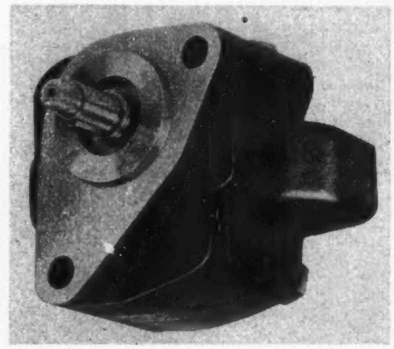
Multi-Pole Relay



MIDGET alternating-current multipole relay with shunt coil for electronic application is announced by R-B-M Division, Essex Wire Corp., Logansport, Ind. Standard voltages range from 1½ to 220 volts, 60 cycle. Maximum noninductive contact rating is 3 amp 24 volts ac, 1 amp 110 volts ac. Contacts are available in from one to four poles, normally-open, normally-closed or double-break. The relay is also available in single-pole normally-open double-break contact, 10 amp rating at 115 volts. Approximate overall dimensions are 2 in. long, 1½-in. high and 1½-in wide.

Hydraulic Pump

GEAR-TYPE hydraulic pump known as Model H657-A has recently been announced by the Aro Equipment Corp., Bryan, Ohio. The unit produces pressures up to 2000 psi, delivering 5½ gpm at 2800 rpm. Volumetric efficiency of the pump is above 90 per cent, while overall efficiency of the pump above 500 psi is 85 per cent or better. The unit is said to be suitable for hydraulic systems demanding pressures from 100 to 2000 psi, such as hydraulic presses, machine tool feed controls, hydraulic jacks, etc. It employs



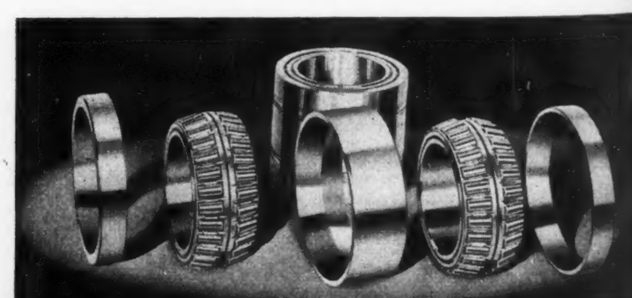
Nitralloy gears in a cast iron body and provides ample porting to standard pipe sizes. Shape and size approximate a 3½-in. cube with ½-in. diameter by 11/16-in. drive shaft extension.

Improved Flow Meter

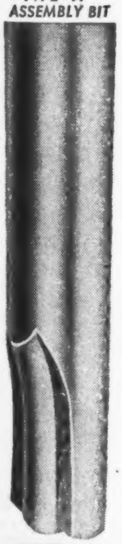
ROTAMETER type of flow meter manufactured by Fischer & Porter Co., Dept. one R-6, Hatboro, Pa., has recently been improved by the use of three equally spaced ribs formed in the tapered glass metering tube. The new Bead Guide is said to eliminate the necessity for guide wire or guide rod formerly used. Numerous difficulties were presented by the old guide wire construction, such as a need for supporting plates, tightening nuts, bushings, etc. Guiding is now done by the precision bore glass tube itself. Glass surface reduces the guiding friction to the vanishing point and a minimum of five metal parts are eliminated.

Taper Roller Bearings

TAPER roller bearings of steel with high nickel content have been announced by Kaydon Engineering Corp., Muskegon, Mich. Of the 4-row type, and of particular value in rolling mills, they are made of a steel alloy containing 3.75 to 4.25 per cent nickel and 1.5 per cent chromium. Heavy shock loads are said to be easily held with this type of steel. Having a deep carburized case, thinner cross sections than ordinarily employed in heavy duty bearings are possible, with the result that roll necks may be larger for a given size mill.



TYPE "A" ASSEMBLY BIT



COMMON SCREWDRIVER

In Power Driving

CLUTCH HEADS OUTMODE ALL OTHER SCREWS

Check This Point by Point Comparison

Regardless of the type of screw you are using now, *RECESSED HEAD OR OTHERWISE,* check their performance against the exclusive CLUTCH HEAD features that short-cut the way to safer, smoother, speedier, and lower-cost production.

✔ High visibility for an easy-to-hit target with resultant increased speed born of operator confidence.

✔ Autcmatic dead-center entry for sure straight driving. CLUTCH HEAD'S Center Pivot Column prevents driver canting.

✔ All-square engagement that eliminates "ride-out" (as set up by tapered driving)...disposing of slippage hazard and need for fatiguing end pressure.

✔ The CLUTCH HEAD Lock-On, uniting screw and bit as a unit for easy one-handed reaching to hard-to-get-at spots.

✔ The unmatched tool economy of the rugged Type "A" Bit which may be restored . . . *reoeatedly* . . . to original efficiency by a 60-second application of the end surface to a grinding wheel.

✔ The only modern screw basically designed for operation with a common screwdriver to simplify field service.

Your request will bring sample screws, Type "A" Bit and illustrated Brochure

This Type "A" Bit is credited by an automobile manufacturer with driving 214,000 screws continuously and without reconditioning . . . on an operation calling for unusually high torque driving.

CLUTCH HEAD SCREWS

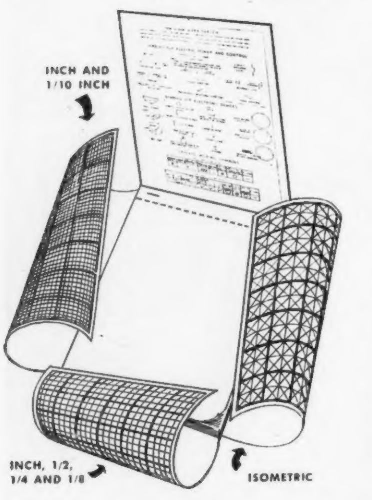
Service men and product users appreciate the convenience of making adjustments with an ordinary screwdriver which need only be reasonably accurate in width . . . the thickness being secondary.

UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 CHICAGO 8 NEW YORK 7

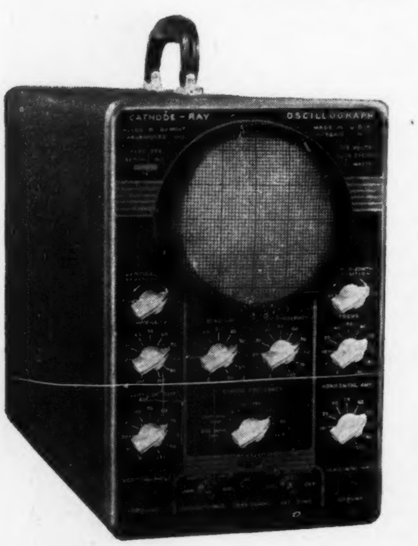
engineering dept equipment

Useful Sketch Pad



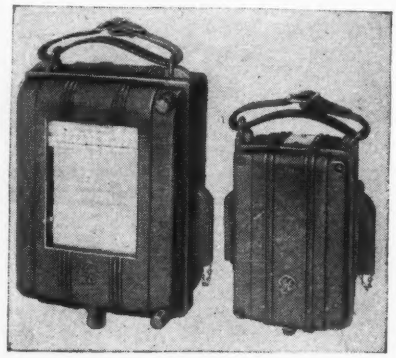
PADS useful to the engineer for sketching machinery while in the shop, or for making tentative drawings have been announced by the Jiffy Sales Co., 1822 E. 37st St., Cleveland 14. The pad provides 75 sheets of 8¾ by 11¼-inch tracing paper held in a folder, three flaps of which provide ruled lines in graph form to guide sketches or lettering. In use, the tracing paper of the "Jiffy Pad" is laid over that cover providing the ruling desired. Lines showing through serve to guide the pencil obviating scales or straight edges, while providing easily reproducible drawings. One cover of the pad is printed with one-inch blocks divided into ⅛-inch squares, a second has one-inch blocks broken down to 1/10-inch divisions, while the third is laid out for sixty-degree isometric sketching with a ⅛-inch rectangular-coordinate background with one-inch major divisions.

Oscillograph



RECENTLY announced is a low cost oscillograph for routine laboratory and production testing. Known as Type 274 the unit is equipped with a 5-inch tube and housed in a steel wrinkled-finished cabinet with a plastic carrying handle. The instrument has a linear time-base with a range of 8 to 30,000 cps, and synchronization may be from vertical amplifiers or external signal. Identical vertical and horizontal amplifiers have a range of 20 to 50,000 cps. Intensity modulation is provided. Sine wave response is uniform within ± 20 per cent from 20 to 50,000 cps and less than 50 per cent at 100,000 cps. Deflection sensitivity: 0.65 rms volt/in.—with amplifiers at full gain, ± 18 rms volts/in. when direct coupled. The cabinet measures 8⅝ in. wide, 14 in. high and 19⅜ in. deep, weighs 35 lb. It is manufactured by Allen B. Du Mont Laboratories Inc., Passaic, N. J.

Temperature Recorder



INKLESS temperature recorder designed for use with resistance temperature detectors has been announced by the Meter and Instrument Division of the General Electric Co., Schenectady, N. Y. Designated as Type CF-2, the unit consists of a portable recording instrument and an external supply unit. It records by means of an inkless recording mechanism on a 4 in. strip chart. Range of fixed scale and chart scale on the recorder is 20 to 140 degrees C. Chart length is 65 ft and the driving motor can be adjusted to provide record length of from eight days to two years on a single chart. Recorder measures 5 by 7¼ by 4 1/16 in., and weighs 11 lb. Accessories available for use with recorder include: Test resistor, leads, and resistance temperature detectors of the bulb, sheath or molded-strip type.

Aluminum Alloy Selector

INFORMATION concerning eighteen of the most widely used aluminum alloys is presented in readily available form in the Reynolds Aluminum Alloy Selector, an 8½ by 11 inch slide-rule chart. For an aluminum alloy under consideration, the chart reads chemical composition of its twelve basic elements, physical constants including specific gravity, density, modulus of elasticity in tension and torsion, coefficient of thermal expansion, thermal and electrical conductivity and specific heat. Data are also given on maximum forging temperature, recommended cycles for solution heat treatment as well as precipitation heat treatment and annealing. For any product form (such as plates or sheets) the chart shows temper available, ranges of tensile strength, yield strength, hardness and elongation and AMS, ASTM, Federal and Navy specifications. Charts are available from Reynolds Metals Co., Department 47, 2500 S. Third St., Louisville 1.

Illustration shows installation of an Alemite LubroMeter Centralized Lubrication System on a conveyor. The system handles either oil or grease—can be designed into almost any type of machine.

THE ALEMITE LubroMeter System is a single line terminating system. It is extremely economical in initial cost, installation and operation.

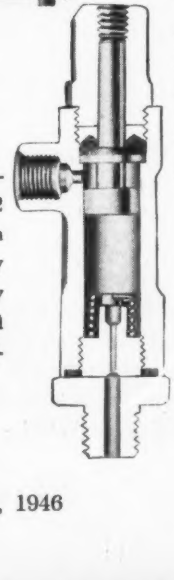
Your machine may have more or less than 22 lubrication points. Even if it has 220, with an Alemite System you introduce lubricants at one point. Push and pull a handle and all points are positively lubricated with a metered quantity of grease or oil without stopping the machines.

Alemite Systems are giving perfect service in every kind of industry on practically all types of machines. They're ending human error, guaranteeing positive lubrication without time-outs . . . eliminating bearing failures . . . lengthening machine life.

Have an Alemite Lubrication Specialist demonstrate one or all 4 new Alemite Systems right at your desk with transparent working models. Also, ask for any technical help you want regarding the Systems. Drop a note on your letterhead to Alemite, 1804 Diversey Parkway, Chicago 14, Illinois.

Cutaway View of Alemite LubroMeter Feeder Valve

Fully hydraulic—Can be mounted directly in bearings—Only 2 moving parts—Indicator on each valve—Lubricant delivery easily adjustable—Valve is of heavy duty construction with hardened steel piston. Available in a variety of sizes and types.



ALEMITE

REG. U.S. PAT. OFF.

Alemite ALONE Combines all 3 in Lubrication

1. EQUIPMENT 2. PROCEDURES 3. LUBRICANTS

Alemite Helps the World Go 'Round

Another Product of STEWART WARNER

MEN... of machines

John W. Greve, since 1939 associate editor of MACHINE DESIGN, has been promoted to the newly created position of managing editor in recognition of "his meritorious work in recent years as associated editor". Before becoming associated with the editorial staff of MACHINE DESIGN, Mr. Greve spent ten years with Westinghouse Electric Corporation where he was responsible for technical writing on mechanical and electrical design. He joined Westinghouse immediately upon graduation from Carnegie Institute of Technology in 1929. Mr. Greve is active in the Cleveland Engineering Society, serving as chairman of the machine design division in 1944, member of the board of trustees in 1945-48, and vice president in 1946.

B. R. Teree, well known in aviation circles, has joined the Weatherhead Co. as project engineer in charge of aircraft development. In his previous connection—with Curtiss-Wright Corp. Airplane Division, Buffalo, from 1931 to January, 1946—he had been in charge of all experimental and production design of hydraulics, landing gears, tail wheel and arresting gears. Also for several years he had been assistant chief of the structures department for the aircraft company. His society activities are many and varied: He has been a chairman of the National Aircraft Standards Sub-Committee on Hydraulics and Pneumatic Systems and Installations, and is now a chairman of the SAE Committee A-6, Aircraft Hydraulic Equipment.

Rear Admiral Lawrence B. Richardson has retired recently as deputy and assistant chief of the Navy Bureau of Aeronautics after twenty-nine years of active naval service, to become associated with Curtiss-Wright Corp. as executive assistant to the president. Because of his background in design, development, manufacture and inspection of aircraft, Admiral Richardson is well fitted for his new responsibilities.

After being commissioned Ensign and serving sea duty in World War I, he completed a postgraduate course at M.I.T. He then became naval inspection representative at various aircraft plants, and later received flight training, qualifying in 1925 as pilot of all types of naval aircraft. During the following ten years he served in various engineering, administrative and industrial capacities in naval aviation for the Bureau of Aeronautics. In 1935 for three years he was connected with the Naval Aircraft Factory, Philadelphia, engaged in producing both engines and airplanes. He returned to the Bureau in 1938 as director of procurement, production and inspection and it was during this period the Navy's largest aircraft pro-

duction program was begun. His promotion to the rank of Rear Admiral came in 1943 when he became assistant chief, Bureau of Aeronautics. Admiral Richardson is a member of the National Advisory Committee for Aeronautics and during the war served as sole Navy representative on the Aircraft Production Board of the WPB.

♦

BJARNE THULIN has returned to the Continental Can Co., Chicago, as design engineer in the equipment development division.

♦

CLAYTON O. DOHRENWEND has resigned as director of the department of mechanics, Illinois Institute of Technology, Chicago, to join Midwest Research Institute, Kansas City, Mo.

♦

JOHN O. FORSTER is the new chief engineer at Aircraft Screw Products Co., Long Island City, N. Y. He formerly held the same post at Bulova Watch Co., Fuel Injection Div., Valley Stream, L.I., N. Y.

♦

SEELEY E. BUCK, formerly project coordinator in the engineering department of Aeronca Aircraft Co., Middletown, O., has become experimental engineer with American LaFrance Foamite Co., Elmira, N. Y.

♦

JACKSON G. KUHN has joined the Los Angeles Scientific Instrument Co. as engineer. He previously had been chief engineer of Gladden Products Corp., Chicago.

♦

THOMAS E. MURPHY has joined the faculty of University of Minnesota as assistant professor of mechanical engineering, soon after leaving the Navy.

♦

GLENN L. MORRIS is the new plant and equipment design engineer at Carnation Co., Oconomowoc, Wis.

♦

WESLEY A. HARPER has become agricultural engineer of Rome Plow Co., Cedartown, Ga.

♦

JOHN HICKENLOOPER, until recently chief test engineer, Wright Aeronautical Corp., Lockland, O., has joined the Philco Corp. refrigeration test laboratory, Philadelphia, as junior engineer.

♦

HUGH J. MULVEY has taken a position as assistant chief of structures with the Piasecki Helicopter Co., Sharon Hill, Pa. He previously had been structural research engineer with Kellett Aircraft Corp., Upper Darby, Pa.

♦

CHESTER E. BECK has been transferred from the Kansas City branch of General Motors Corp. as coach service engineer, to the GMC Truck and Coach Division in Pontiac, Mich., as air conditioning and heating engineer.

♦

VINCENT JOSEPH ZARDUS has been promoted from layout draftsman to transmission design engineer with Kellett Aircraft Corp., North Wales, Pa.

♦

MAJOR WALTER H. NILES, who formerly had been connected with USAAF and is now a partner in the industrial design firm of Bruce Kamp Associates, New York and Philadelphia, recently was awarded the Legion of Merit in presentation ceremonies at Mitchel Field, New York, for his work in developing automatic controls for the eventual "pushbutton" airplane.

♦

F. R. MCLEOD has been appointed assistant chief engineer in charge of construction and maintenance activities of canning, quick freezing and plant facilities, Deerfield Packing Corp., Bridgeton, N. J.

♦

G. L. REYNOLDS, an automotive engineer, has been appointed chief engineer of Novo Engine Co., Lansing, Mich.

♦

ALBERT L. PRESTON has been assigned to the Oak Ridge, Tenn., plant of Fredric Flader Inc., as senior mechanical engineer, doing experimental and development work on gas turbines, jets, and other high performance powerplants.

♦

C. WILLIAM BOLLER has resigned as designer with Pratt & Whitney Aircraft Division of United Aircraft Corp., to become engineering manager of Pine-Ihrig Machine Co., Oshkosh, Wis.

♦

ROBERT C. HENN has returned from the services to his post as product design engineer with Monroe Calculating Machine Co., Orange, N. J.

♦

THOMAS JAY REEVES is development engineer with Continental Motors Corp., Muskegon, Mich. He has just been released from the Army.

♦

WILLIAM R. HUMES has left the Douglas Aircraft Co. where he served as design engineer to accept a position in the same capacity with North American Aircraft Co., Inglewood, Calif.

♦

WILLIAM I. TRUETTNER has returned to the mechanical engineering department of the Agricultural and Mechanical College of Texas, College Station, Tex., after spending 18 months as a member of the faculty of the University of Puerto Rico as an exchange professor.

♦

WILLIAM W. THOMSON, formerly experimental engine draftsman, Dymaxion Dwelling Machines Inc., Wichita, Kans.. has been made experimental development engineer, Wilson Zuck Aircraft Co., Tulsa, Okla.

♦

DALE L. BENNETT has become connected with Kiekhaefer Corp., Cedarburg, Wis., as chief engineer. He had previously served in this capacity with Kropp Forge Co., Chicago.

♦

DR. STEWART G. FLETCHER has been engaged by Latrobe Electric Steel Co. as chief research metallurgist.

♦

W. M. WALWORTH has succeeded L. C. JOSEPHS JR, as vice president and chief engineer, Mack Mfg. Corp., New York City. Mr. Walworth joined the company in 1939 and has been acting chief engineer of the corporation since March, 1945.

♦

FRANK T. CHRISTIAN has been made chief engineer of the Eclipse Machine division of Bendix Aviation Corp. Associated with the Eclipse engineering staff since 1929, Mr. Christian has since 1940 directed engineering for the

corporation's commercial production of the Bendix starter drives and bicycle coaster brakes.

♦

William T. Stephens, who previously had been executive vice president of Hydraulic Control Engineering Co. and vice president in charge of engineering and sales of the Hydraulic Equipment Co., has been elected president and chief engineer of Industrial Hydraulics Corp., Cleveland.

♦

G. Eldon Irving has been made chief engineer of the Venderlator Mfg. Co., Fresno, Calif.

♦

Eugene Laas is chief pump engineer of Chandler-Evans, division of Niles-Bement-Pond Co., West Hartford, Conn. He had formerly held the same position with Chandler Evans Corp., South Meriden, Conn.

♦

Paul H. Winter has been appointed chief engineer of Hahn Motors Inc., Hamburg, Pa. He had formerly been connected with Rogers Diesel & Aircraft Corp.

♦

N. E. Risk in addition to his supervision of all engineering and co-ordinated activity pertaining to the application of allied equipment and special attachments produced by Caterpillar Tractor Co., will assume charge of transmission design.

♦

Gordon C. Seavey, formerly research engineer with Air-cooled Motors Inc., Syracuse, N. Y., has been appointed chief mechanical engineer, Ultrasonic Corp., Boston.

♦

Frank H. Erdman has become a design engineer for McDonnell Aircraft Corp., St. Louis. He resigned as an engineer from Wright Aeronautical Corp., Paterson, N. J.

♦

Robert M. Hollowell has returned from the service and is now with the Republic Supply Co. of Calif. as a designing engineer on oil field machinery.

♦

Major-Gen. G. M. Barnes, recently resigned deputy chief of Ordnance and chief of the Ordnance Department's Research and Development Division, has been made engineering assistant to Edward G. Budd, president, Edward G. Budd Mfg. Co., Philadelphia.

Criticizes Welding Specifications

To the Editor:

Arc welding has been of decisive importance to America. It did more than most other manufacturing processes in the production of tools of war during World War II, and has produced a record for reliability in billions of welds, made over many years, that is unmatched by any other manufacturing process.

In spite of this splendid record, arc welding is being attacked in a way that is tremendously handicapping its application, and promises to interfere with its future use still more. This attack is aimed not at the process, as such. Obviously, such tactics would fail. It is, rather, a technique of throwing suspicion on the process by writing into specifications expensive and impractical tests which have little to do with the excellence of the weld. Most of these tests have to do with infinitesimal variations of no possible importance, but of great cost. This attack already has eliminated economic use in many proper applications and, if continued, soon will eliminate many others.

We see, for instance, the ruling that welds must be x-rayed, which increases the cost by several times, yet the commercially welded joint is always of greater strength than the parent metal and is tremendously stronger than any riveted joint, for which x-raying never has been suggested.

We see riveted joints which are made tight by caulking, a process which is accepted without question. The resulting undercut is enormous, yet a welding undercut that is extremely small frequently is cited as the reason for rejection of welds.

We see welds chipped out, rewelded, and welded vessels rejected because of trifling defects such as infinitesimal porosity either on the surface or beneath, yet more serious defects in the parent metal of the same structure, the weakening effects of which would be tremendously greater, are accepted without question.

Again, we see welding electrode specifications being written which enormously increase the cost of production with no increase in either the reliability nor in the excellence of electrodes. Rivets have no such test to handicap them. While welding electrodes are tested in every conceivable and nonsensical way, no one suggests any tests for rivets, yet riveted joints are always the weakest spot in any structure. This is never true of a full-sized welded joint.

Much time and expense are expended in testing electrode deposit to make sure it has great ductility, and the weld is rejected if the ductility is low, yet riveted joints have no elongation and are accepted without question. The contour of a weld deposit is a matter of very close inspection, yet no one examines the contour of any rivet or the hole it only partially fills.

Because of the higher elastic limit of the weld metal, there is no load that can be put on a welded structure, in which the weld is of equal or greater section than the parent metal, which can affect the weld in any possible way until great distortion of the rest of the structure has taken place. Such distortion would make that structure valueless for its intended purpose, yet all this testing and rejecting listed in the foregoing is made mandatory in many welded structures—never in riveted structures. Further instances of the same kind can be cited by the score. The examples shown, however, are sufficient for the writer's purpose.

Welding over the years has done a more reliable job than the rivets it has replaced. That record is conclusive. The engineering profession, which relies so completely on welding in so many cases, must recognize and resist this studied attempt to eliminate the arc welding process. As has been indicated, the attack already has eliminated the economic use of welding in many structures.

The success of such an attack on this tremendously valuable method is neither good advertising for the engineering profession nor good ethics for those involved in the attack. It is time we dealt with reality.

—J. F. Lincoln, *President*
The Lincoln Electric Co.

Mechanical Seals

(*Concluded from Page* 142)

sibility of cocking. Asking the machinist to pull up the bolts so that the angle between the two flanges is zero (within the narrow limits good sealing requires) is asking a great deal, even with a torque wrench. To ask him to do it every time he works on the machine is practically asking the impossible. This difficulty might be avoided, as mentioned earlier, by the use of an O-ring, as shown in *Fig.* 28.

It is preferred that the ring be installed in this manner, with the step in the lower flange. If this step is omitted, as shown in *Fig.* 29, there might be trouble in bringing the two pieces together without pinching the O-ring. These rings should be installed with a diametral squeeze or press, that is, the radial dimension of the cavity confining the ring should be less than the free cross section diameter of the ring. Moreover, for best results the corners into which the O-ring is deformed by the fluid pressure should be square or with a small radius.

Leading Edge Is Chamfered

If the leading edge of the O-ring counterbore in the upper piece in *Fig.* 29 were not chamfered it might spoil the O-ring as it came down due to the diametral squeeze. On the other hand, if it were chamfered one of the corners into which the O-ring seats would have a radius with an external center, which is not conducive to good sealing. It is possible, of course, to chamfer the edge of the seal bearing surface, and to bring the two pieces together with the O-ring in the upper part rather than the lower, but the necessity for the machinist's remembering to do it that way rather than the reverse is eliminated by the design in *Fig.* 28, where the leading edge of the O-ring counterbore can be chamfered widely without producing an external radius in the corner, and the parts can be assembled either way.

Another advantage in the use of O-rings is the possibility of reducing weight, an important feature at the end of the machinist's day. The upper seal stationary member in *Fig.* 1, for example, need not be in the form of a flange, with heavy bolting, as in the lower member, because it is unnecessary to provide gasket pressure. The saving is doubled, since the companion flange is also eliminated. The lower member is gasketed because the unit is designed for possible use in fluids or vapors harmful to O-rings. The O-rings as shown seal nothing but the oil between the upper and lower seals.

Considering the Machinist

In nearly every plant there is at least one machinist who, confronted with a departure from the method he learned when he served his apprenticeship years ago, treats it as a temporary procedure, unwelcome, and doomed before trial. These men are often valuable for their caution and experience, although their tendency to resist change can become annoying. They may be without authority but they are not without power. More often than not they are ingenious and, if one is able to instill in them respect for a new idea, their help, sometimes out of the line of duty, may mean the difference between success and failure. During the war, when plants employed large numbers of these old-timers, some equipment manufacturers found it not only advantageous but frequently necessary to show them more courtesy than had been their custom.

Such steps probably come under the heading of public relations, but the engineering department can assist a great deal by designing with the idea in mind that the skepticism mentioned is by no means limited to the older men. A young machinist may approach a new machine element with an open mind. He is willing to give the gadget a fair trial. But he finds that in order to remove a setscrew it is necessary to dismantle most of the machine. Some nuts can be unscrewed only a sixth of a turn for each wrench hold.

Special tools are often required to reach around corners and into deep cavities, and he doesn't have them. There is no way to lift a heavy flange except by wedging it progressively higher from side to side until two men can get a grip under it. After he has scraped his knuckles, bruised his fingers, strained his neck, and stubbed his toe, he begins to lose patience. He is convinced that the person who designed the machine should be made to work on it. He begins to visualize a college graduate with no practical experience, who designs these miserable contraptions by day and plays bridge with the general manager by night. He buys the old-timer a beer after work and decides that the old stuffing box wasn't such a bad design after all.

Building Morale on the Drawing Board

Any industrial value that the seal had is now seriously jeopardized. The discouraged machinist will talk to his friends, and before long any work on the seal will be classified in the category of things to be avoided. It is surprising what an obstacle such a situation can be when the machine manufacturer's representative calls at the plant to correct mechanical trouble. If the designer had but gone through each assembly step, and had conceived the operations in three dimensions at the start, he might have prevented a great deal of the difficulty by making small changes here and there to facilitate the work. The suggestion is applicable, of course, to the designing of any machine, but it is particularly applicable to the designing of machines in which mechanical seals are to be used. In many cases they are new to the men who must work with them, and first impressions gained from simple pleasure and pain reactions might create or preclude the all-important respect of the men. It is a matter of building morale on the drawing board, and its importance for a new machine element such as a seal should never be underestimated.

The concluding article in this series will discuss theoretical design of mechanical seals.

PHOTOGRAPHS may be reproduced on metal, wood, cloth, leather, plastics or almost any other surface by a new process using an emulsion developed by the Glenn L. Martin Co. The emulsion is spread on the material to be sensitized and, after exposure, can be developed similarly to the commercial process.

Business and Sales Briefs

PREVIOUSLY executive vice president, William W. Gilmore has succeeded Walter B. Schulte as president of the Micro Switch Division of First Industrial Corp., Freeport, Ill.

◆

Establishment of a new Hydraulics Division to specialize in the development of hydraulic equipment has been announced by Rockwell Mfg. Co., Pittsburgh. W. H. Marsh will serve as general manager.

◆

Associated with the company for twenty-five years, Herbert C. Petzing has been appointed to the post of manager of the Cleveland branch office of Ahlberg Bearing Co., Chicago. He has replaced Max Palmer who recently started a bearing distributor business under the name of Palmer Bearing Co. and will carry a full line of Ahlberg bearings.

◆

National Malleable & Steel Castings Co., Cleveland, has elected Cleve H. Pomeroy as president to succeed Charles H. McCrea who died recently. In addition to acting as president, Mr. Pomeroy will continue to hold the office of treasurer.

◆

With William V. Pyndus in charge, a new office has been opened at 416 Loew Theatre Bldg., 108 West Jefferson St., Syracuse, N. Y., by the Penflex Sales Co. division of Pennsylvania Flexible Metallic Tubing Co. Mr. Pyndus will serve territory formerly handled by four other branch offices. In addition to Syracuse, his territory includes such cities as Binghamton, Buffalo, Elmira, Jamestown, Niagara Falls, Rochester, Rome, Schenectady and Utica, as well as many smaller manufacturing centers.

◆

Richard F. Muller has been promoted from assistant manager to manager of the New Orleans district office of Allis-Chalmers Mfg. Co., Milwaukee. He succeeds F. W. Stevens who will continue with the company as a special representative in the New Orleans office. Mr. Stevens has been associated with Allis-Chalmers for forty-one years.

◆

A new Boston district office has been established at 38 Newbury St. by the Ward Leonard Electric Co. of Mt. Vernon, N. Y. Kasson Howe, previously with the sales engineering department in the home office, has been named district manager.

◆

Progressive Welder Co. has appointed K. P. Swanson, 204 Chapel St., Abington, Mass., to serve as its representative in eastern Connecticut, eastern Massachusetts, Rhode Island, Maine, Vermont and New Hampshire. Mr. Swanson joins V. A. Chern and H. B. Axtell, who have offices at 15 Gramercy Park, New York 3, in covering the New England area for the company's line of resistance welding equipment. Mr.

Chern handles the New York metropolitan district as well as western Connecticut and western Massachusetts, while Mr. Axtell covers eastern New York and northern New Jersey.

♦

Associated with steel sales work in the New York and New England district since 1938, J. W. Burdick has been made sales manager of the New York district of Washington Steel Corp., Washington, Pa. He will maintain a temporary sales office in Hartford, Conn.

♦

Recently announced is the appointment of P. L. Coddington as manager of sales for the Welded Alloy Tube Division of The Carpenter Steel Co., Kenilworth, N. J. Serving as assistant manager of the division will be J. A. Dietrich.

♦

With headquarters at 1700 North Kostner Ave., Chicago 39, B. M. Loewenstein has been named general sales manager of Howard Foundry Company. He will head the sales activities of the company's four plants which produce aluminum, brass, bronze, magnesium and semisteel castings.

♦

Three new regional representatives have been appointed by The Lindsay Corp. They are: Reinhofer & Breaux, 222 West Adams St., Chicago, to cover the Central states; Lindsay-Pacific Co., 607 Hearst Bldg., San Francisco, to serve the Pacific Coast; and Thomas T. Tucker, 1102 Candler Bldg., Atlanta, to cover the southeastern region. Offices and plant of The Lindsay Corp. are now located in a modern new building at 1740 Twenty-fifth Ave., Melrose Park, Ill.

♦

Aetna Rubber Co. has elected James W. Kelley as president to succeed Charles Mashek who died recently. Mr. Kelley is also board chairman of the J. W. Kelley Co., producer of industrial oils.

♦

Change of name to Newark Plastics Inc. has been announced by Newark Plastic Engineering Co. of Newark, O., manufacturer of molds and special tools for the plastic industry. M. W. Burkhart is president of the new corporation, which is affiliated with Nu-Engineering Inc. of Ferndale, Mich. Both companies offer a complete engineering and manufacturing service for all types of molds and special tools and fixtures used in plastic manufacturing.

♦

Formerly sales engineer for the special products department, J. W. LeRoy has been appointed manager of special products sales for the Berger Mfg. Division of Republic Steel Corp., Canton, O.

♦

Acquired recently by Clarostat Mfg. Co. Inc. of Brooklyn, N. Y., is the entire outstanding stock of Kurman Electronics Corp. with offices, research laboratory and plant in Long Island City, N. Y. The company will be operated as a wholly-owned subsidiary but with its own engineering and production personnel and plant. Victor Mucher is the new president of the subsidiary.

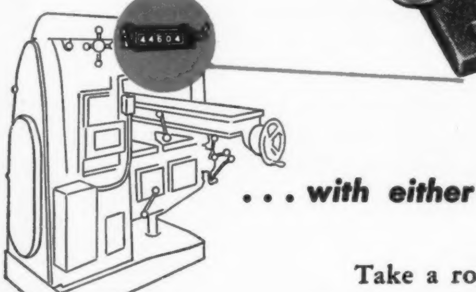
♦

Steel Improvement & Forge Co., Cleveland, has named Carl I. Schweizer to head the newly established Turbine Forgings Division which will specialize in parts of high

You Can COUNT ON VEEDER-ROOT

to Build "COUNTROL"

into Your Product or Process



. . . with either Standard or Special Veeder-Root Counting Devices

Take a rolling mill, a machine tool, jet plane, juke box — or what have *you?* Odds are long that there's a place ready and waiting, *though perhaps as yet undiscovered*, for Veeder-Root *Count*rol in your product-design and sales-promotion plans.

And now's the time to find out. Ask a "Counting House" engineer to count up the ways in which you can profit by building into your product or process the right Veeder-Root Device (standard or special) . . . like, for one instance, the standard Medium Size Rotary Ratchet Counter shown above.

Remember, too, that *Count*rol can profit your customers as well as yourself . . . can help to keep you on friendly terms by proving your product's guarantee. Find out just how much you *can* count on. Write today to Veeder-Root Inc., Hartford 2, Conn. *In Canada:* Veeder-Root of Canada, Ltd., 955 St. James St., Montreal 3. *In England:* Veeder-Root Ltd., Dickinson Works, 20 Purley Way, Croydon, Surrey.

Give your product this extra Merchandising Feature

temperature alloys required for gas turbines and turbosuperchargers. For the last three years Mr. Schweizer has spent the major part of his time in working out methods for forging the newest alloys. Also announced by the company is the appointment of Arthur Zimmerman as sales manager. Formerly Mr. Zimmerman was associated with the Weatherhead Co., Cleveland.

♦

Acquisition of the Brake Equipment & Supply Co. of Chicago has been announced by H. K. Porter Co. Inc. The new plant manufactures brake parts for railway locomotives and freight cars, and airbrakes for export.

♦

Formerly assistant to the president, Charles Rinderle Jr. has been elected treasurer and secretary of Eastern Air Devices Inc., Brooklyn, N. Y., manufacturer of electric motors and electrical specialties.

♦

David C. Crowley, 531 Esperson Bldg., Houston 2, Texas, has been named representative in Texas for Colonial Broach Co. and Colonial Bushings Inc. He has a broad background of service and sales engineering work in the metal working field.

♦

Following the resignation of Henry W. Jackson, Alfons Alven has been elected president of the Bearings Co. of America, Lancaster, Pa. In the early spring Mr. Alven was elected a director of the company, which he joined in 1932 as district manager of the Chicago office. He is a graduate of the Stevens Institute of Technology in mechanical engineering.

♦

Harvey P. Barnes, 345 East 218th St., Euclid, O., has been named sales engineer for the Cleveland and northern Ohio territory of Hannifin Mfg. Co., Chicago 24, manufacturer of pneumatic and hydraulic production equipment.

♦

Connected with the company since 1912, William J. Norman has been appointed district manager of the Waverly plant of United States Steel Supply Co., Newark, N. J. He succeeds Charles Kramer who retired recently after forty-six years of service.

♦

Bundy Tubing Co., world's largest manufacturer of small metal tubing, has announced that six administrative departments have been moved from Hern Ave. to new general offices on East Jefferson Ave. at Parker, Detroit. The main plant of the company is on Nine Mile Road and a research laboratory is nearing completion at Coolidge Highway & Maple Road.

♦

With offices at 405 Union Central Bldg., Cincinnati, Gray L. Furey has been named sales representative for the Kentucky, southwestern Ohio and eastern Indiana territory of Titan Metal Mfg. Co., Bellefonte, Pa.

♦

Allegheny Ludlum Steel Corp. has consolidated its sales development and engineering service divisions under the managership of W. B. Pierce, who was formerly manager of the sales development division. The new department will co-ordinate and extend the company's co-operation with users and

fabricators of stainless steel on their problems of applications and uses, and will pay special attention to the development of new markets for the introduction of new alloys.

Recently announced is the formation of the Taco West Corp., 2620 South Park Ave., Chicago. The company will manufacture and market automatic electronic control devices in the fields of combustion control, gas analysis, pyrometry, process control and allied apparatus. President of the new concern is Theodore A. Cohen, who founded and was formerly vice president of the Wheelco Instruments Co.

Simplex Engineering Co., Zanesville, O., has appointed J. H. Stewart of Barton, Vt., as New England representative for the company's high pressure hydraulic pumps and valves.

Meetings and Expositions

Nov. 14-15—
American Management Association. Special production conference to be held at Palmer House, Chicago. Henry J. Howlett, 330 West 42nd St., New York, is secretary.

Nov. 17-22—
American Welding Society. Twenty-seventh annual meeting to be held at Hotel Ambassador, Atlantic City, N. J. M. M. Kelly, 33 West 39th St., New York 18, is secretary.

Nov. 18-22—
National Metal Congress and Exposition to be held at Municipal Auditorium, Atlantic City, N. J., in conjunction with the annual meetings of the following societies: American Society for Metals; Iron and Steel Division and Institute of Metals Division of American Institute of Mining and Metallurgical Engineers; American Welding Society; and American Industrial Radium and X-Ray Society. Chester L. Wells, 7301 Euclid Ave., Cleveland 3, is assistant managing director of the exposition.

Dec. 2-4—
Society of Automotive Engineers Inc. Air transport engineering meeting to be held at Edgewater Beach Hotel, Chicago 14. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

Dec. 2-6—
American Society of Mechanical Engineers. Annual meeting to be held in New York City. Additional information may be obtained from headquarters of the society at 29 West 39th St., New York 18. C. E. Davies is secretary.

Dec. 2-7—
Seventeenth National Exposition of Power and Mechanical Engineering to be held at Grand Central Palace, New York. Charles F. Roth, at the same address, is manager of the exposition.

Dec. 9-11—
Society for Experimental Stress Analysis. Annual meeting to be held at Hotel New Yorker, New York. Additional information may be obtained from headquarters of the society at P. O. Box 168, Cambridge 39, Mass. W. M. Murray is secretary-treasurer.

Jan. 6-10—
Society of Automotive Engineers Inc. Annual meeting and engineering display to be held at Book-Cadillac Hotel, Detroit. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

Jan. 14-17—
National Materials Handling Exposition to be held at Public Auditorium, Cleveland. Additional information may be obtained from Clapp & Poliak Inc., 37 Wall St., New York.

Jan. 23-26—
Society of the Plastics Industry. Second conference and exhibit of the Low-Pressure Division to be held at Edgewater Beach Hotel, Chicago. W. T. Cruse, 295 Madison Ave., New York, is executive vice president of the society.

Jan. 27-31—
American Society of Heating and Ventilating Engineers. Fifty-third annual meeting to be held in conjunction with seventh international heating and ventilating exposition at Lakeside Hall, Cleveland. Charles F. Roth, Grand Central Palace, New York, is manager of the exposition.

Jan. 28-29—
National Warm Air Heating & Air Conditioning Association. Thirty-third annual meeting to be held at Hotel Cleveland, Cleveland. George Boeddener, 145 Public Square, Cleveland 14, is managing director.